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# SLIDE-VALVES.

A BOOK FOR PRACTICAL MEN

ON THE

PRINCIPLES AND METHODS OF DESIGN;

WITH

*AN EXPLANATION OF THE PRINCIPLES OF SHAFT-GOVERNORS.*

BY

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# PREFACE.

A SERIES of articles entitled "Valve-Gears" was published by the author in *Power* during 1895 and 1896. The object of these papers was to put the principles of design of slide-valves in a practical form for practical men. This book contains the subject-matter of those papers entirely revised and rearranged, with a complete new set of cuts.

C. W. MACCORD JR.

AUBURN, N. Y.,
July 4, 1897.

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# SLIDE-VALVES.

## CHAPTER I.

### GENERAL PRINCIPLES.

FIG. 1 shows a cylinder which is entirely empty, except for the piston shown; the latter being free to slide in the cylinder under the influence of any force. Suppose, for

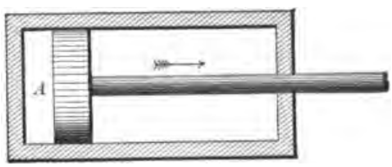



FIG. 1

example, that steam were admitted to the space *A* at the left. The natural result would be that the piston would move over to the right-hand end, as shown in Fig. 2. Now,

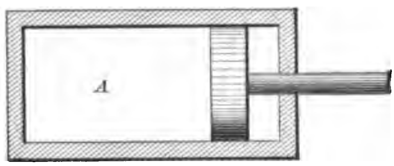



FIG. 2

in order to bring the piston back to the position shown in Fig. 1, a force must be applied to the right of the piston.

Suppose steam were admitted there. This would not suffice to move the piston, because the space $A$ is filled with steam and the pressures on the two sides of the piston would balance each other. It therefore follows that the mere admission of steam to the ends of the cylinder in alternation is not enough to cause the piston to reciprocate, or move backward and forward; but, in addition, some means must be provided for emptying or exhausting the steam from the cylinder as soon as it has done its work. If this exhausting is accomplished, the steam-pressure will then move the piston in the desired way.

In Fig. 3 is represented the simplest contrivance designed to regulate the admission and exhaust of steam from the

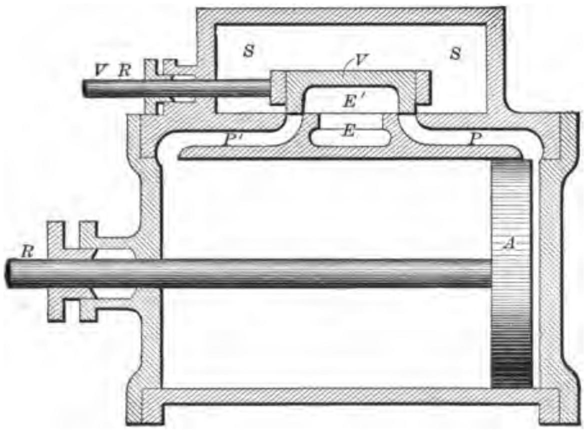



FIG. 3

cylinder. The space $S$ is called the "steam-chest," and is in direct communication with the boiler, so that it is at all times full of steam at boiler-pressure. The passages $P$ and $P'$ lead to the right- and left-hand ends, respectively, of the cylinder, and are called "steam-ports" or, more often, "ports." The piece $V$ is the "valve," and it is free to slide in a horizontal direction only, or parallel with the " line of

stroke" of the piston. The surface of the main casting in which the ports terminate is called the "valve-seat," and the lower surface of the valve, which is in contact with the seat, is called the "valve-face." The valve is moved by means of the "valve-rod," *VR*, which passes through a "stuffing-box" at the left of the steam-chest; the object of the stuffing-box being to prevent the leakage of steam. The piston, *A*, with its "piston-rod," does not require any particular description. The passage *E* in the valve-seat is called the "exhaust-port," and it is in direct communication with the atmosphere at all times.

The valve as shown is separated entirely frcm the "valve-gear" which operates it. This is done because a clearer understanding of the subject can be had by first studying thoroughly the movements of the valve and then considering the gear required to produce these movements.

The operation of this valve is as follows:

In Fig. 3 the valve just covers both ports, and the steam cannot reach either end of the cylinder. Then the valve

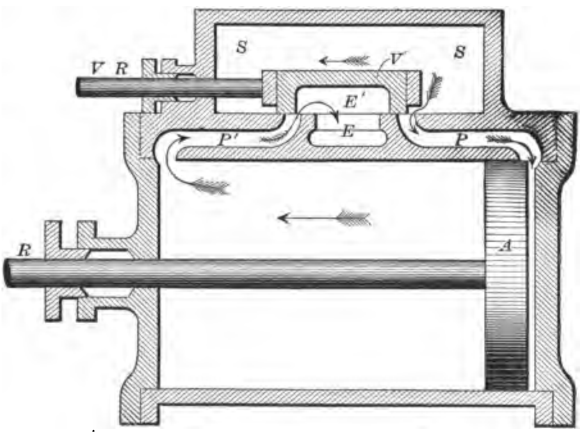



FIG. 4

moves slightly to the left, assuming the position shown in Fig. 4, when the right-hand port is uncovered, and the steam

naturally follows the course indicated by the curved arrows, thus getting on the right of the piston and forcing it over to the left, as shown by the straight arrow. When the piston reaches the middle of its stroke the valve is at the extreme left of its stroke, or travel, and begins to move in the opposition direction; this being shown in Fig. 5. When the piston reaches the extreme left-hand position the valve is

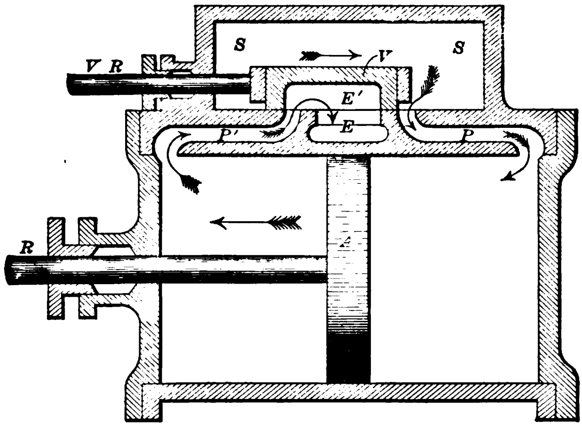



FIG. 5

again "central" or in the middle position, and just covers both ports, as shown in Fig. 6. The valve continues to move to the right, thus admitting steam to the left-hand end of the cylinder, as shown in Fig. 7. At the same time, it will be noticed, the cavity $E'$ in the valve is over the right-hand steam-port, thus permitting steam to escape from the right-hand side of the piston through the exhaust-port $E$, as shown by the arrows. The exhaust-port leads to the atmosphere. The piston now reverses its stroke, and when in the middle of its stroke the valve is over at the extreme right-hand end of its travel, as shown in Fig. 8. The piston continues to move to the right, while the valve reverses and moves to the left, and the original condition, shown in

Fig. 3, is once more attained. The relative movements of the piston and valve are then repeated. When the valve is

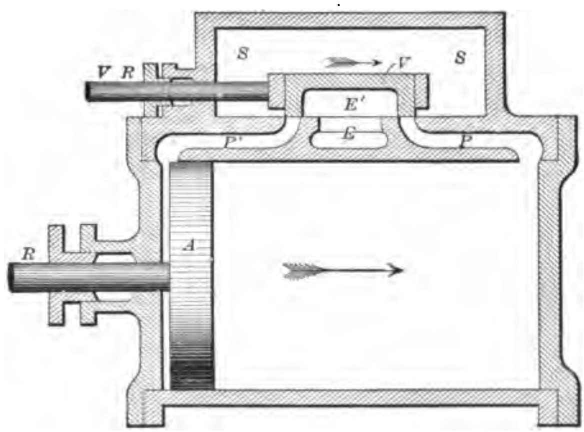



FIG. 6

again over at the left, as shown in Figs. 4 and 5, the steam at the left of the piston is exhausted through *P'* and *E*.

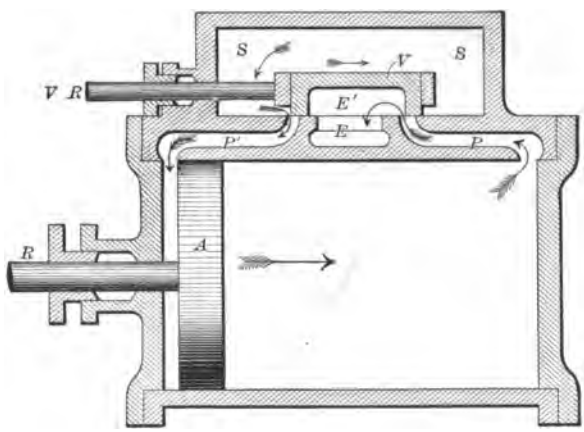



FIG. 7

It should be at once apparent that the motion of the valve bears a fixed relation to the motion of the piston, and should

be derived from it in some way. When the valve is in its central position the piston is at either one end or the other of

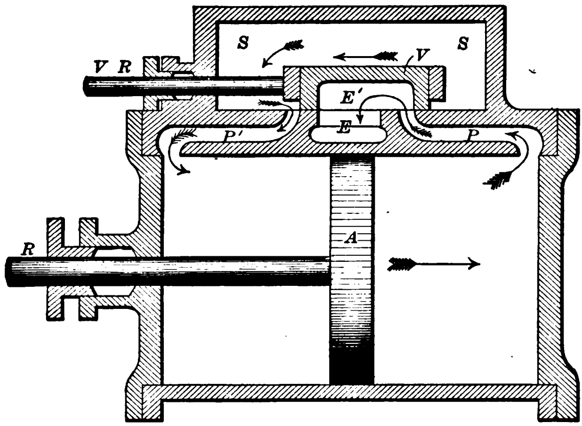



FIG. 8

its stroke. That is, the valve is either a half-stroke ahead of or behind the piston.

In every engine the reciprocating or "to-and-fro" motion of the piston is converted into a rotary motion of the fly-wheel. Fig. 9 shows the simplest possible contrivance for effecting this conversion; and at the same time it shows the simplest form of apparatus for deriving the motion of the valve from the stroke of the piston. The center of the crank-shaft, on which the fly-wheel is keyed, is at *O*, and the crank-pin is at *C*. The crank itself is omitted for the sake of clearness. The piston-rod, *R*, terminates in a slotted cross-head, in which the crank-pin fits and is free to slide. With this gear, as the piston moves to and fro, the crank rotates, and the crank-pin slides up and down in the slotted cross-head. It will be noticed that when the positions of the piston-rod, crank-pin, and shaft are as shown in the figure—that is, when their centers are all in the same straight line,—there is no rotative effort on the crank-pin, the entire pressure on the

piston being expended in an attempt to crush the crank-pin or shaft. At the other end of the stroke a similar state of affairs occurs. These two points are called the "dead-points" or "dead-centers." If an engine, when stopped, gets on either dead-center, it is necessary to pry it off before

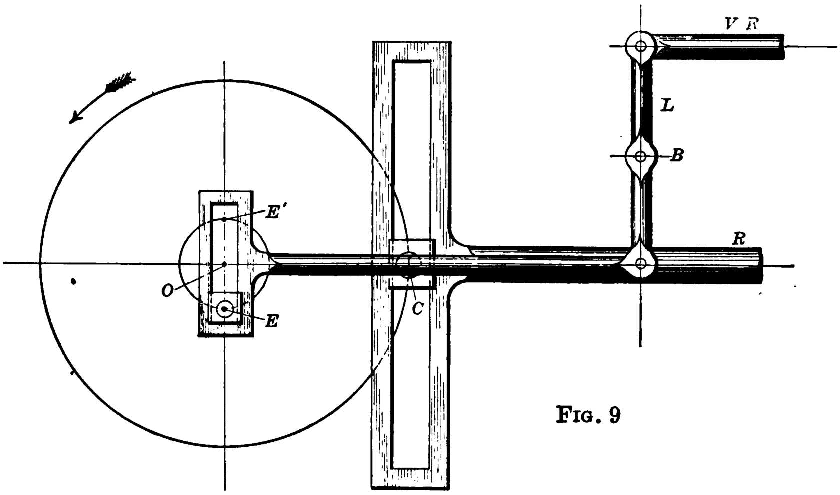



FIG. 9

it can be started. When the engine is on the center the piston is at the end of its stroke.

Another pin is secured to the shaft just 90°, or a quarter circumference, away from *C*, as shown at *E*. If, therefore, the valve be driven by *E*, it will always be a half-stroke away from the piston. The pin *E* is called the "eccentric-pin," or the "eccentric," and another slotted cross-head is employed to communicate the motion of *E* to the valve-rod, *VR*, through the "rocker," or "rock-shaft," *L*. This rocker is pivoted at *B*, which is some point on the frame of the engine; and, in the case shown, the arms are equal, so that the motion of the valve is the same in amount as if the valve-rod were fastened to the end of the eccentric-rod; but it is opposite in direction, or the rocker reverses the move-

ment of the valve. A rocker is not an indispensable adjunct, but it is used when the line of travel of the valve is above or below the line of stroke of the engine. The type shown in the cut is only one of many, some of which will be described later, and is taken on account of its simplicity.

This device answers the two requirements mentioned; for the motion of the valve is obviously derived from the stroke of the piston, and, as the angle between the crank and the eccentric is 90°, it is evident that when the piston is at one end of its stroke, as it must be in Fig. 9, the rocker *L* is vertical and the valve is consequently in the middle position.

The distance *OE* from the center of the crank-shaft to the center of the eccentric-pin is called the "throw" of the eccentric, or simply the "eccentricity," and a momentary inspection of the figure will show that the travel of the valve is equal to twice the eccentricity, or to the diameter of the circle in which the eccentric-pin travels. Similarly, the length of the crank, *OC*, is often called the "throw" of the crank, and it is equal to one-half the stroke of the engine. It will be noticed that the travel of the valve is much less than that of the piston. It is made so in order to decrease the power required to move the valve; for the friction between the valve and the valve-seat varies directly with the distance through which the valve has to be moved.

This type of valve-gear has two serious drawbacks: The steam is not used expansively, and there is no "compression" or cushioning of the steam at the end of the stroke to take up the momentum of the moving parts, to avoid "pounding." Both of these faults are due to the fact that steam is admitted to the cylinder from the moment that the piston leaves one end of the cylinder until it reaches the other end, or, in other words, "steam follows full stroke," a fact which should be plain from the foregoing text. The exhaust is also open during the entire stroke.

If steam were "cut off" from the cylinder some time

before the piston reached the end of its stroke, the expansive force of the steam would, in combination with the reduction of pressure in front of the piston due to the exhaust, carry the piston along to the end of the stroke. There is great gain in economy in so using steam expansively, but it lies without the province of this work to enter upon its proof. Similarly, if the port were closed to exhaust a short time before the piston reached the end of its stroke, a certain volume of steam would be imprisoned and compressed by the advancing piston, thus acting as a cushion to overcome the momentum of the moving parts.

Steam may be cut off from the cylinder at any desired point in the stroke of the piston by adding to the outside of the valve a piece equal to the distance that the valve has moved from its central position when the piston is at that point of the stroke. This is shown in Fig. 10, which is a

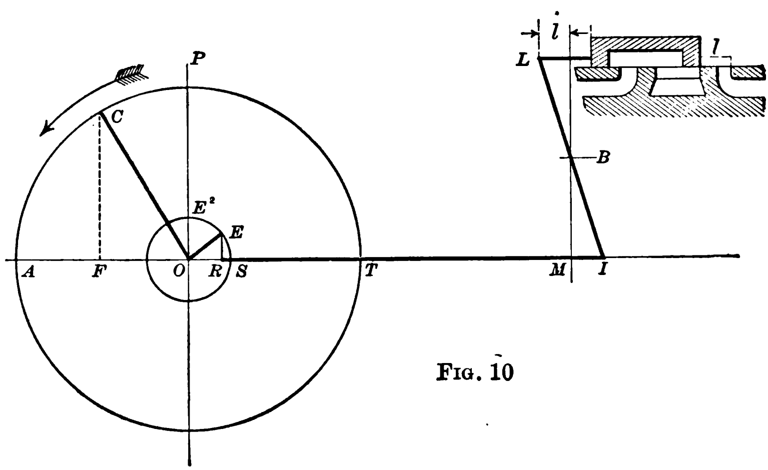



FIG. 10

center-line sketch of the valve-gear shown in Fig. 9, with the addition of the valve and ports at the end of the valve-rod. Suppose, for example, that it is desired to cut off at three-fourths stroke—that is, when the piston has completed three

fourths of its travel toward one end of the cylinder the port shall be closed to steam.

FIRST. *Find the position of the crank-pin when the piston is at the point where cut-off is required.* This is done by describing, about $O$ as a center, a circle with a radius equal to the length of the crank; then take $TF$ equal to $\frac{3}{4}$ of $AT$, and draw a perpendicular to $AT$ from $F$ until it cuts the crank-circle at $C$, the required crank-pin position; and $OC$ is the "crank-line," or line representing the position of the center-line of the crank. This is true because, with the slotted connection, the horizontal movement of the crank-pin is the same as that of the piston.

SECOND. *Locate the eccentric-pin position corresponding to the determined crank-pin position.* This is done by drawing $OE$, the eccentric line, perpendicular to $OC$, because the angle between the crank and eccentric is 90°, and describing a circle about $O$ as a center, and with a radius equal to the eccentricity. This circle cuts $OE$ at $E$, which is the required position of the eccentric-pin.

THIRD. *Locate the position of the end of the eccentric-rod.* To do this, it is only necessary to drop a perpendicular $ED$ from $E$ to the line of stroke.

FOURTH. *Draw in the rest of the figure.* This is simply done by taking the dimensions of the valve-rod, eccentric-rod, rocker, etc., from Fig. 9.

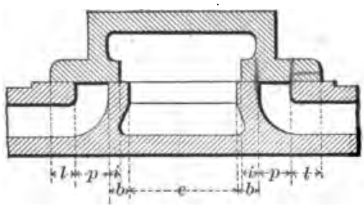
Now, the "displacement of the valve," or the distance it has moved from its central position, is $OR$ or $l$, and it is evident that the valve must move that distance to the right before the right-hand port is closed. If a piece of the length $l$ is added to the outside of the valve, as shown by the dotted lines, the port will be closed with the gear in the position indicated, and the desired result will be obtained for the right-hand end of the stroke.

The angle which the crank-line at any time makes with the line of stroke is called the "crank-angle"; and with the

slotted connection it is evident that for the same piston position on the forward and return strokes the crank-angles are equal. Consequently the valve displacements are equal, and, taking the case just solved, it is only necessary to lengthen the valve on the other end by the amount $l$ in order to secure cut-off at the same point on the return stroke.

The end of the cylinder nearest the crank is called the "crank end," the other being called the "head end." The stroke in which the piston approaches the crank-shaft is called the "forward stroke"; and the stroke in which it recedes from the crank-shaft is called the "return stroke."

The length $l$ which is added is called the "outside lap" or "steam-lap," because it is on the outside of the valve and controls the admission of live steam. It is defined as the amount which the valve projects over the outside edge of the steam-port when the valve is in its central position, as shown in Fig. 11. It is very evident that if the valve originally just



*l, l,* = *outside laps.*
*i, i,* = *inside laps.*
*b, b,* = *bridges.*
*e,* = *exhaust port.*
*p, p,* = *steam ports.*

FIG. 11

covered the port when in its central position, a piece of length $l$ would project just that amount over the edge of the port.

The compression of steam may be secured by adding a length $i$ to the inside of the valve, the value of $i$ being determined in the same manner that $l$ was found; that is, by making it equal to the distance which the valve would have to travel in order to close the port, with the piston in the

required position. This addition is called the "inside lap" or "exhaust-lap," for obvious reasons. It may be defined as the amount which the valve projects over the inside edge of the port, with the valve central, as in Fig. 11.

The partition which divides the exhaust from the steam-port is called the bridge. The inside lap therefore rests on the bridge.

A difficulty due to the introduction of outside lap is at once apparent. The admission of steam is made late by just the amount $l$. That is, the valve must move a distance $l$ from its middle position before the port is opened to steam, or the eccentric must turn through the corresponding angle. This may be remedied by changing the angle between the crank and eccentric, moving the eccentric ahead—that is, in the direction in which the engine is to run—an angle corresponding to $Z$: That is, in Fig. 10, through an angle $EOP$ making the eccentric position $OE^2$. This will make the admission occur sooner, but it will also make the cut-off occur earlier, so that the result will be different from the three-quarter cut-off desired, as shown in Fig. 12, where $E^2$ is the eccentric position with the port closed, and $OC$ the corresponding crank-position. It is not necessary to show how this three-quarter cut-off could be obtained, as the general method has been indicated, and this type of gear is only useful as a mode of explanation. By increasing the angle between the crank and eccentric, the opening of the port to exhaust will be made earlier—that is, it will occur before the end of the stroke, in the simple valve with outside lap but no inside lap; and the compression will occur before the end of the return stroke. The inside lap, if added, need be much smaller than the outside lap. Early exhaust opening, provided it does not occur earlier than seven-eighths stroke, does no harm, as it simply serves to reduce the pressure against which the piston must work on its return stroke.

The angle between the crank and the eccentric has now

been changed from 90° to $COE^2$ by the advance of the eccentric, and the amount of the change is called the "angular advance"; and angular advance is therefore defined as the

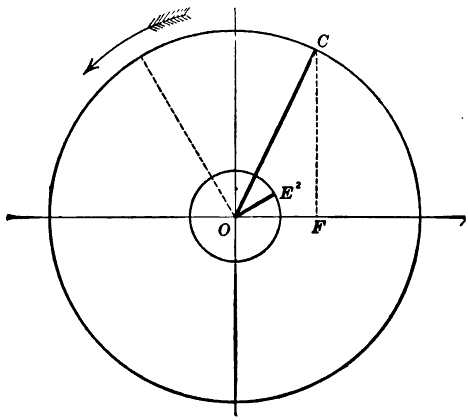



FIG. 12

difference between 90° and the angle between the crank and eccentric.

The rocker affects very materially the angle between the crank and eccentric. With the device shown, the horizontal motion of the upper arm is directly opposite to that of the lower arm. If, in the simple device shown in Fig. 9, the eccentric-rod were fastened direct to the valve-rod, and the rocker in consequence eliminated, it would be necessary to move the eccentric-pin around to $E'$, which is 180° away from $E$, in order to secure the same movement of the valve. If, with the eccentric in this new position, outside lap were added to the valve, it would be necessary to move the eccentric ahead—in the direction in which the engine is to run—to secure the admission of steam at the beginning of the stroke, just as before. But in this case it would tend to *increase* the angle between the crank and eccentric, while with the rocker it resulted in a decrease of the angle. In the case of

direct connection the eccentric precedes the crank, while with the reversing rocker the eccentric follows. The definition of angular advance may therefore be stated as follows:

"*Angular advance*" is the difference between the angle between the crank and eccentric and 90°.

When the right angle is the greater the angle becomes the "angle of follow," or the eccentric follows the crank.

When the right angle is the less, the angle becomes the "angle of advance," or the eccentric precedes the crank.

This affords a ready means of determining whether a rocker is used.

With this type of valve-gear the valve displacement for a given piston position is found by drawing the perpendicular from the eccentric-pin to the line of stroke, as *ER*, Fig. 10. The displacement from the end of its travel is *RS*, and from the central position is *OR*. The latter is the one usually employed. It must be remembered that, on the contrary, "piston displacement" is the amount the piston has moved from the beginning of its stroke, and is usually expressed by the fraction of the stroke completed.

It has been pointed out that setting the eccentric ahead makes all the events of the stroke earlier. This may be carried to such an extreme as to reverse the direction of rotation of the crank. This may be explained by the use of Fig. 13.

Let *CO* be the crank and *OE* the eccentric, with the engine on the head-end dead-center, running in the direction shown by the arrow. The relative positions of crank and eccentric are taken from Fig. 12. At this stage the valve would commence to move to the left, opening the head-end port. But suppose that the eccentric-pin were suddenly slipped around to $E^1$, the angle $COE^1$ being equal to *COE*. The valve displacement would remain the same, so that the right-hand port would be just covered. But if the engine continues to move in the direction of the arrow, the valve will move to the right, with the eccentric at $E^1$, thus opening

the right-hand or head-end port to exhaust and the crank-end port to steam. The difference between this motion of the valve and that obtained with the eccentric at $E$ is shown in the figure. The piston is moving toward the left in both cases; but with the head-end port exhausting and the crank-

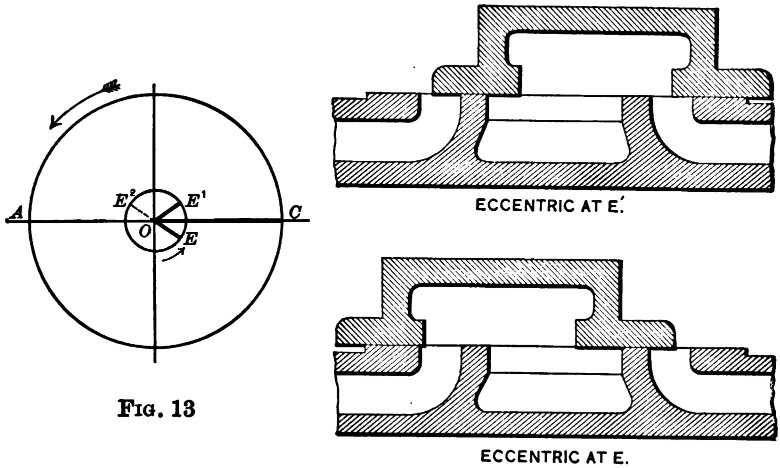



FIG. 13

end port admitting steam, as shown by the upper valve, there soon comes a time when the pressure at the left of the piston overcomes the momentum of the moving parts and forces the piston to reverse its movement. This carries the crank in the opposite direction. The engine will continue to run in this reversed direction, as $E^1$ is as much behind the crank in the new motion as $E$ was behind the old.

It will be noticed that in the first case, with the eccentric at $E$, the crank goes above the line of stroke on the forward stroke. The engine is said to "run over." When reversed, the crank goes below the line of stroke on the forward stroke, and the engine "runs under."

The results obtained from Fig. 13 may be stated as follows:

*To reverse an engine*, it is only necessary to slip the eccentric-pin around on the shaft until it is on the other side of the crank-pin, making the angular distance between the crank and eccentric the same that it was at first.

In this connection it is to be noted that it is not correct to slip the eccentric around 180° on the shaft, until it is directly opposite its original position. This condition is shown by $OE^2$, and it is at once apparent that that will make all the events of the stroke late by the angle $E^1OE^2$. Of course, if the angle between the crank and eccentric were 90°, it would be correct to slip it around 180°, and that is the only case in which it would be right.

during the last half the crank-pin moves from $C$ to $M$. With a uniform velocity of the crank-pin, which is assumed, the piston travels slower during the last half of the forward stroke

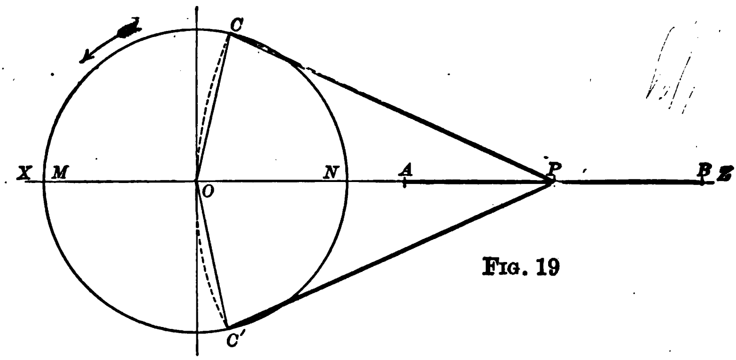



FIG. 19

than during the first half. On the return stroke, with the piston at $P$, the crank position is $C'O$, and the crank angle is $C'OM$, which is evidently greater than $CON$. It is also

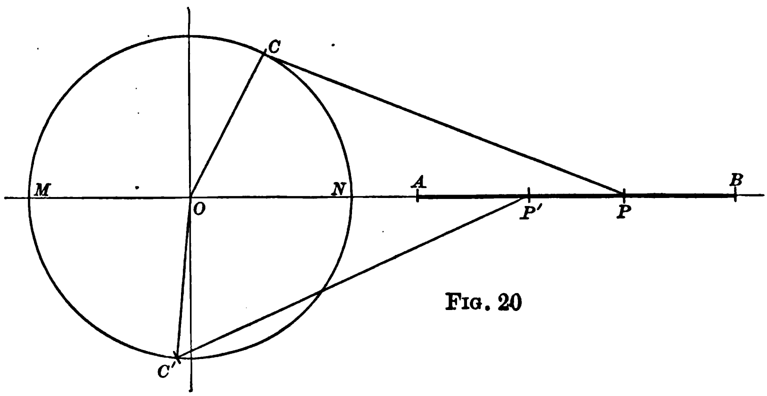



FIG. 20

evident that the piston velocity is less during the first half of the return stroke than during the last half; for the crank-pin, moving at a uniform rate, takes longer to pass over $MC'$ than it does to cover $C'N$.

In Fig. 20 the piston-displacements are the same as in Fig. 18, but the crank-pin positions are found by striking arcs about $P$ and $P'$ as centers, the radius in each case being the length of the connecting-rod. The crank angles $C'OM$ and $CON$ are evidently unequal, $C'OM$ being the greater.

The foregoing may be summed up as follows:

*With the ordinary crank and connecting-rod the crank angles corresponding to equal piston displacements on the forward and return strokes are unequal, that on the return stroke being the greater.*

*With the ordinary crank and connecting-rod, assuming the velocity of the crank-pin to be uniform, the piston velocity is not uniform, being less during the crank-end half of each stroke than during the head-end half.*

The variation between the crank angles on the forward and return strokes depends on the length of the connecting-rod as compared with that of the crank; the longer the connecting-rod, the less will be the difference between the crank angles for equal piston displacements on the forward and return strokes. When the rod is infinitely longer than the crank, there is no difference between the angles; and hence the slotted-cross-head form of connection is usually referred to as the "infinite connecting-rod." With the ordinary types of engines, where the rod is from four to ten times the length of the crank, the difference must be considered, and affects the amounts of lap required on the ends of the valve to secure cut-off at the same point in each stroke, making the laps unequal. This is because the angle between the crank and eccentric is fixed, and the valve displacement varies with the crank-angle; hence if the crank-angles are unequal, the valve displacements are unequal; and the lap required to cut off at any point in the stroke being equal to the valve displacement at that point, it follows that unequal crank-angles require unequal laps.

With the eccentric the case is different, as the eccentricity

is so much less than the length of the eccentric-rod that the effect of angularity may be neglected. This is because the eccentricity is much less than the throw of the crank, while the eccentric-rod is equal to or longer than the connecting-rod; hence the ratio of the eccentric-rod to the eccentricity is much greater than the ratio of the connecting-rod to the crank. The valve displacement is therefore reckoned as equal to the eccentric displacement.

It is, therefore, necessary to determine the crank positions corresponding to the piston positions at which cut-off is desired, when designing a valve for any engine; or, when studying the action of a valve already in use, to determine the piston position corresponding to a given crank position. The method of Figs. 19 and 20 may be employed, but it has the disadvantage of requiring to be drawn on a large scale, and of having the lines intersect at acute angles, making it hard to obtain good results. A better method is shown in Fig. 21. This method is a modification of the original method of M. Marcel Deprez, and the author acknowledges his indebtedness to the *Practical Engineer*.

Lay off the straight line $XY$ of indefinite length, and on it take any convenient point, as $O$, for a center; and about $O$ describe a circle with a radius equal to the length of the crank. This gives the crank-circle, whose diameter is $MN$. Lay off the distance $OO'$, equal to the square of the crank length, divided by four times the length of the rod, or

$$OO' = \frac{C^2}{4R},$$

where $R =$ length of connecting-rod;

$C =$ length of crank.

About $O'$ as a center, and with a radius equal to the length of the crank, describe an auxiliary crank-circle, whose diameter is $M'N'$. The point $O'$ should always be between

the center of the crank-shaft, $O$, and the cylinder; that is, in the case shown, the cylinder is at the right.

Now determine the crank-pin position corresponding to the given piston position on the *auxiliary* crank-circle, as if the slotted cross-head were used. For example, if the piston is at $\frac{1}{8}$ stroke, take $N'D$ equal to $\frac{1}{8}$ of $N'M'$, and draw $Dc$

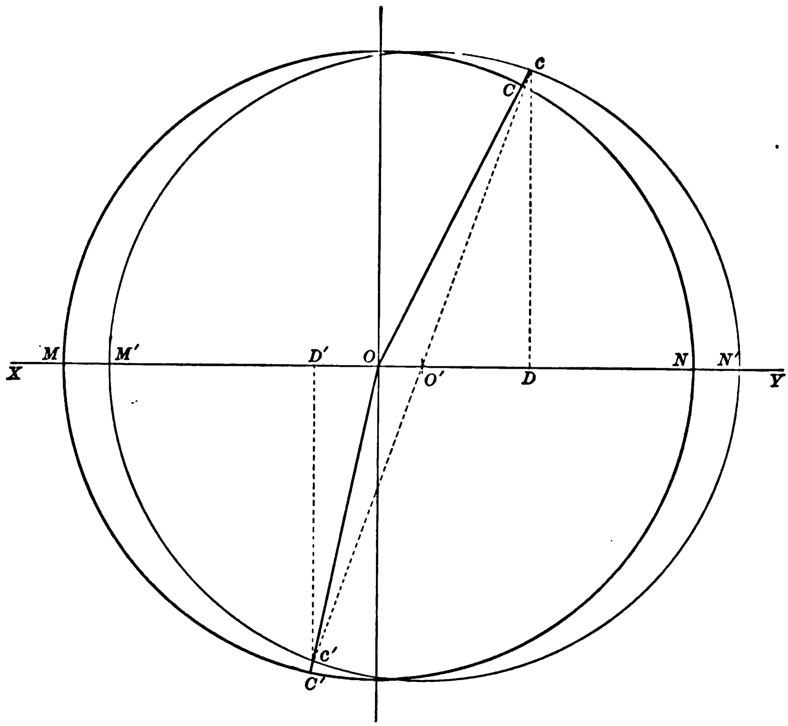



Fig. 21

perpendicular to $XY$ until it cuts the auxiliary circle at $c$. Then $c$ is the crank-pin position on the auxiliary circle.

Next, from $c$, draw a line to $O$. Then the part of this line $cO$ included within the crank circle is the crank position corresponding to the given piston position; that is, $CO$ is the required crank position. This diagram shows very clearly the

difference between the finite and infinite rods. For $cO'$ is the crank position corresponding to the given piston position with the infinite rod, and the crank angle $cO'N'$ is evidently different from $CON$.

For the return stroke the process is similar. Take $M'D'$ equal to the given piston displacement. Draw $D'c'$, perpendicular to the line of stroke until it cuts the *auxiliary* crank-circle at $c'$. Join $c'$ and $O$, and produce $Oc'$ until it strikes the crank-circle at $C'$. Then is $OC'$ the crank position corresponding to the given piston displacement.

A reversal of this method will give the piston displacement corresponding to any given crank position. Draw the given crank position, and determine its intersection with the *auxiliary* circle, producing the given crank-line, if necessary. From this intersection drop a perpendicular upon the line of stroke. The distance from the foot of this perpendicular to the dead-center of the *auxiliary* crank-circle gives the piston displacement.

It will be seen at once that the distance $OO'$ varies with the ratio of the connecting-rod to the crank, and that the difference between the crank angles depends on the length of $OO'$.

To save calculation, Table I has been prepared, which gives the values of $OO'$ in terms of the crank length. For example, if the stroke length is 36 inches and the connecting-rod is 5 feet 3 inches long, the distance $OO'$ is $1\frac{9}{32}$ inches, if laid out full-size, found as follows: The crank is one half of the stroke, or 18 inches, and, the connecting-rod being 5 feet 3 inches or 63 inches length, the ratio of the connecting-rod to the crank is $63 \div 18 = 3\frac{1}{2}$. From Table I it is found that for this ratio the distance $OO'$ is equal to the crank length divided by 14 or multiplied by .0714. Using the latter gives $18 \times .0714 = 1.2852$ inches. From Table II it is found that the nearest fraction of an inch to .2852 is

.28125, corresponding to $\frac{9}{32}$ of an inch. $OO'$ is therefore made $1\frac{9}{32}$ inches.

If the diagram is to be made on any other scale, $OO'$ is reduced or enlarged in proportion. At half-size it would be .6426 or $\frac{41}{64}$.

## CHAPTER III.

### VALVE DIAGRAMS—GENERAL PRINCIPLES.

In designing a new valve, or in studying the action of an old one, it is necessary to know the valve displacement corresponding to a given crank position. This may be found by making a center-line sketch of the gear employed, but this requires too much space, is not sufficiently accurate, and is too long and tedious.

There are various diagrams designed to show at a glance the relation between the crank position and the valve displacement, all having certain good points. The one which will be used in this work is that developed by Dr. Gustave Zeuner.

### Case I.

That of a simple valve, with neither outside nor inside lap; angle between crank and eccentric 90°. Such a valve is shown in Figs. 3 to 8.

Draw the two lines $XY$ and $VZ$, Fig. 22, at right angles, intersecting at $O$. This point $O$ represents the center of the crank-shaft, and $VZ$ represents the line of stroke. About $O$ as a center, and with a radius equal to the length of the crank, describe the crank-circle as shown. Take $OE$ on $XY$ equal to one-half the eccentricity, and describe the valve-circle shown. Locate $E'$ in the same way, and draw the lower valve circle.

Now, having this figure, *the valve displacement for any crank position is equal to that portion of the crank line which*

*is included in the valve-circle.* For example, when the crank is in the position $O1$, the valve displacement is $O7$.

This diagram shows the action of the valve to be just as described in Chapter I. Suppose the engine to be running

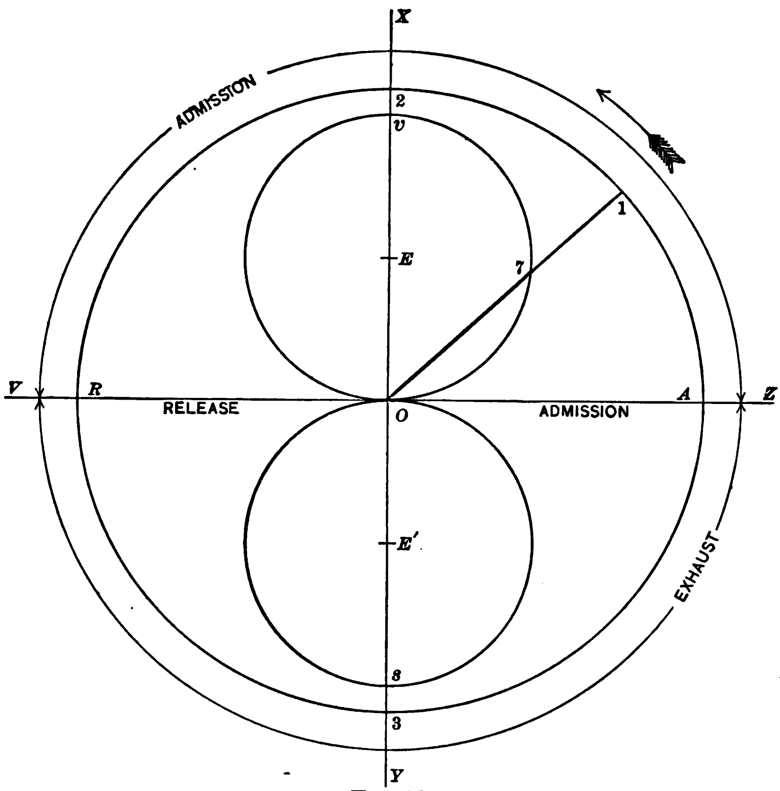


FIG. 22

over, as shown by the arrow; then the action of the valve on the head-end is as follows: With the crank at $A$, the crank line is $AO$, and the valve displacement is zero, as $AO$ does not cut either valve-circle. When the crank moves ahead to 1, the valve displacement is $O7$. When the crank reaches 2,

at right angles to line of stroke, the valve is displaced the distance $Ov$, which is the eccentricity, or half travel. While the crank travels from $A$ to 2 the port is gradually opening. From 2 to $R$ the port gradually closes, until at $R$ it is closed. The valve now commences to reverse its direction, as shown by the fact that the crank line, as it moves on, cuts the lower valve circle. The valve is at the other end of its travel when the crank-pin is at 3, as shown by the fact that the valve displacement is $Os$, equal to the eccentricity. When the crank reaches $OA$, the valve is again central.

With this type of valve the port-opening is equal to the valve displacement. Therefore, during the entire forward stroke, or while the crank-pin moves from $A$ to $R$, the head-end port is open to admission, and during the entire return stroke, or while the crank-pin goes from $C$ to $A$, the head-end port is open to exhaust, as shown in the figure.

The action of the valve on the crank end is directly the reverse; that is, the valve displacement which opens the head-end port closes the port at the crank end. The events of the strokes are therefore read from opposite valve circles, as shown in Fig. 23. With the crank at $A'$ the valve is central, but as it moves on the crank-end port is opened to admit steam, by the amount of the valve displacement, until, with the crank-pin at 3, the crank-end port is wide open. From there on to $R'$ the port is being closed, until at $R'$ the valve is central. Admission and exhaust are therefore reversed.

## Case II.

Simple D-valve, with outside lap; angle between crank and eccentric 90°.

The only difference between such a valve as this and that of Case I is that the port-opening for admission is not equal to the valve displacement, but is less by the amount of outside lap, as pointed out in Chapter I. The exhaust remains

unchanged. The diagram for this case is shown in Figs. 24 and 25.

In order to find the port-opening for a given crank position, it is only necessary to subtract the amount of the lap

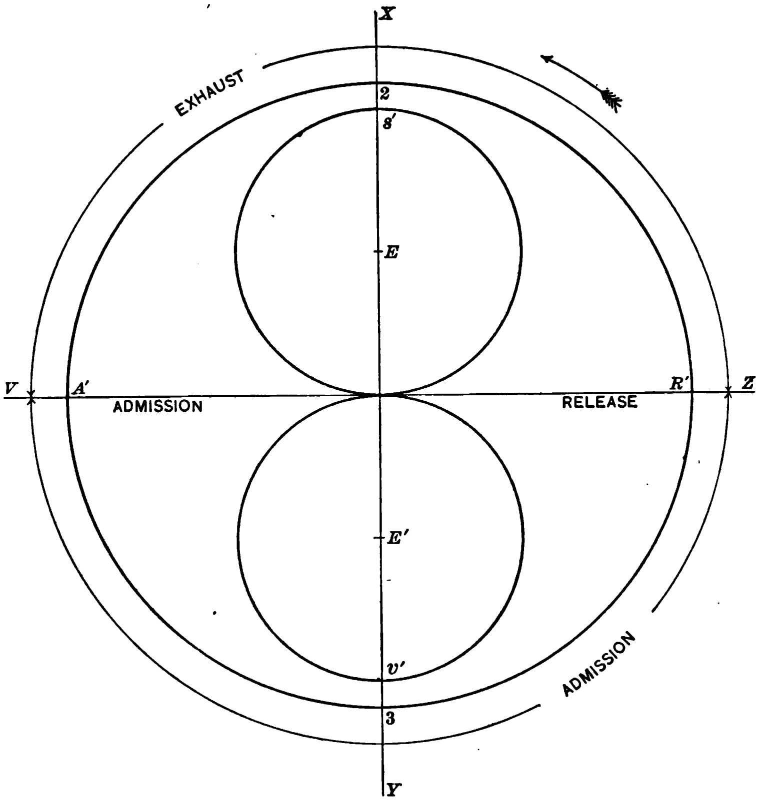


FIG. 23

from the valve displacement; and this can be done by striking an arc about $O$ as a center and with a radius $OL$ equal to the outside lap. This being done, the port-opening is given by the following

RULE: *The port-opening for any given crank position is equal to that portion of the crank-line which is included*

*between the lap and valve-circles.* For example, with the crank-pin at 2, Fig. 24, the port-opening is $Lv$.

Fig. 24 represents the diagram for the head end. Events on the forward stroke are read above the line of stroke;

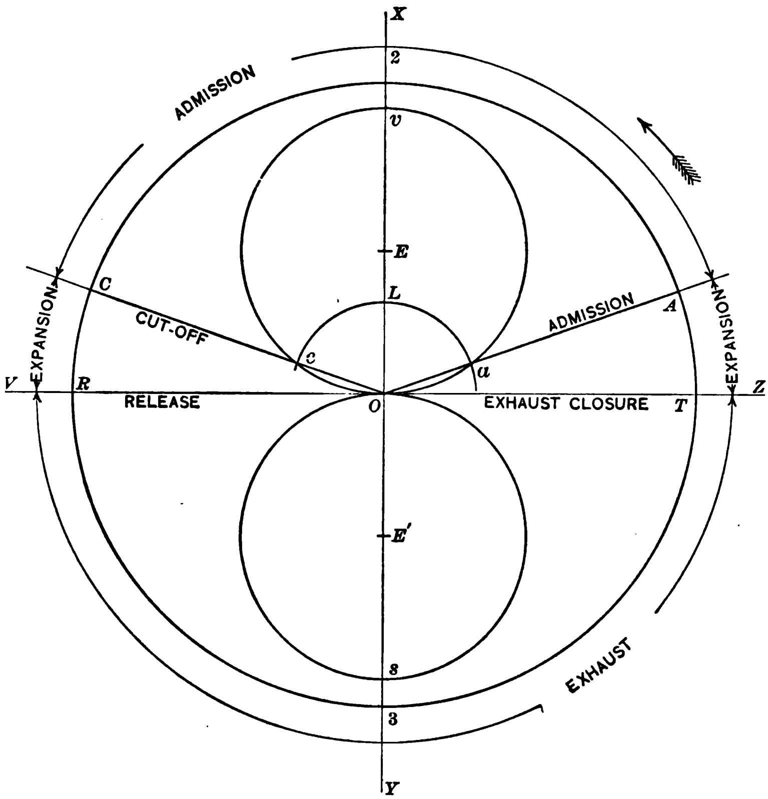



FIG. 24

events on the return stroke are read below it, the direction of rotation of the engine being shown by the arrow. With the crank at $T$ there is no displacement of the valve. When it gets to $A$, the displacement is just equal to the lap, and any further movement of the crank will open the head-end port

to steam. Therefore admission begins at $A$; and it lasts until the port is again closed, which will be when the crank is at $C$, when the valve displacement is equal to the outside lap. This gives the two following rules:

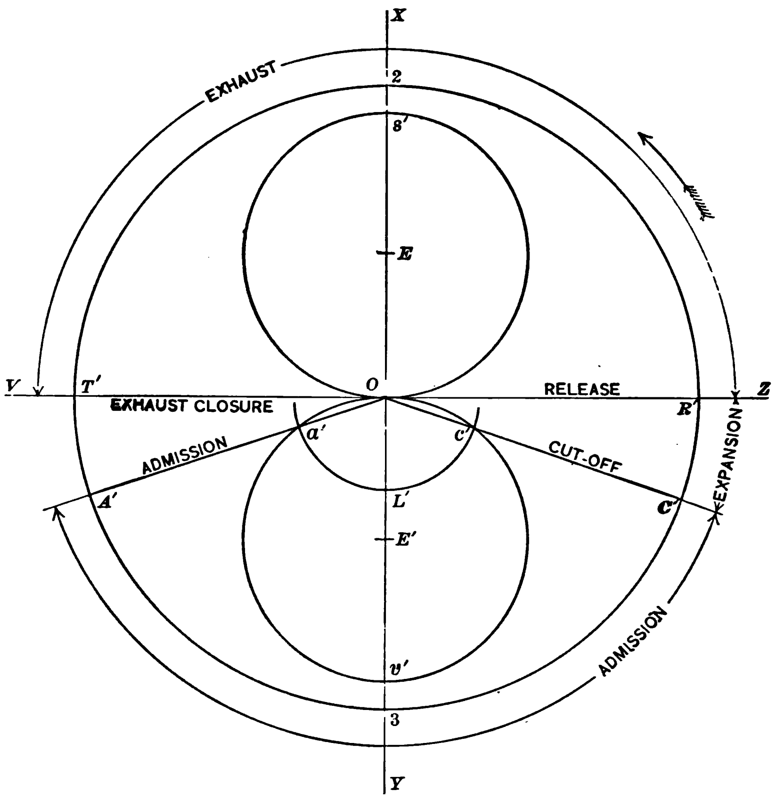



FIG. 25

"*Admission*" *begins when the crank line passes through the first intersection of the valve and outside-lap circles.* $OA$ passes through $Oa$.

"*Cut-off*" *occurs when the crank line passes through the second intersection of the valve and lap circles.*

In Fig. 24 the lap-circle is only drawn above the line of stroke. This is because the outside lap does not affect the exhaust, and the exhaust and admission are read from opposite valve-circles.

The exhaust opens at $R$, and closes at $T$—that is, it lasts during the entire stroke, just as in Case I.

From $C$ to $R$ no steam is admitted, and the steam admitted up to cut-off expands.

From $T$ to $A$ the cylinder is empty, and the engine runs only by virtue of the momentum of the fly-wheel.

Fig. 25 shows the diagram for the crank-end of the cylinder, the valve being supposed to have the same outside lap on both ends.

## CASE III.

Simple D-valve, both outside and inside lap; angle between the crank and eccentric 90°.

This case is shown in Figs. 26 and 27, the former being for the head-end, and the latter for the crank-end. The valve is supposed to have the same amount of inside lap at both ends. The valve travel, crank throw, and outside lap are the same as in the preceding problem.

The inside lap affects only the exhaust; and as its effect is to lessen the amount that the port is open to exhaust by the amount of lap, allowances can be made for it by striking an arc about $O$ as a center, and with a radius $OI$ equal to the inside lap, in a manner similar to that employed for the outside lap. This gives as a rule:

*The port-opening to exhaust corresponding to any given crank position is equal to that portion of the crank-line which is included between the valve-circle and inside lap-circle.* The exhaust-opening and steam-opening are read from different valve circles, it must be remembered.

Fig. 26 is the same as Fig. 24 as far as the facts relating

to admission and cut-off are concerned. The exhaust does not open until the valve displacement is equal to the inside lap, or until the crank is at $R$. It remains open while the

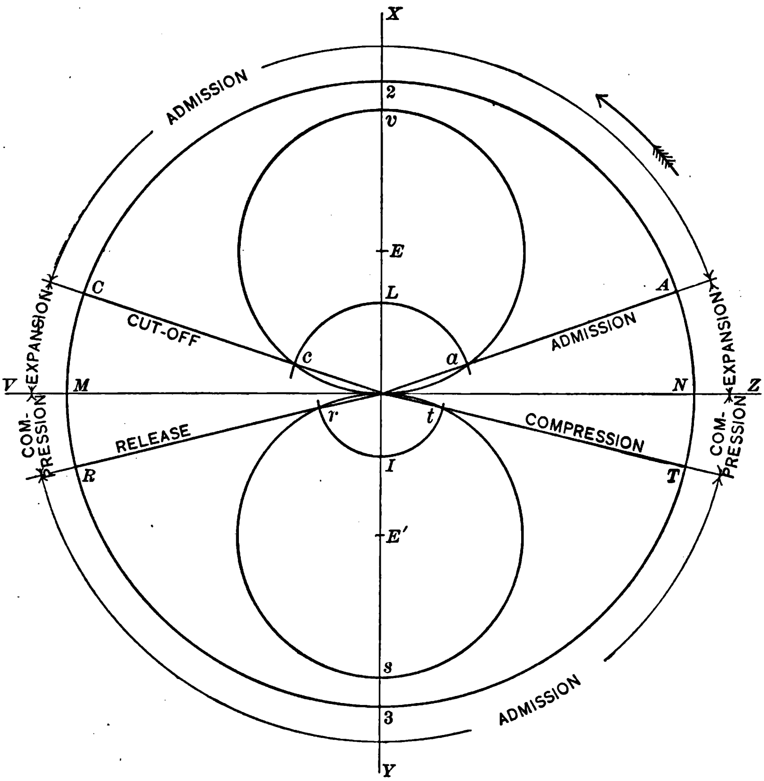



Fig. 26

crank passes from $R$ to $T$, when the valve displacement is again equal to the inside lap. From which these two rules:

*"Release," or exhaust-opening, occurs when the crank-line passes through the first intersection of the valve and inside lap circles.* $OR$ passes through $r$.

*"Compression," or exhaust-closure, occurs when the crank*

*line passes through the second intersection of the valve and inside-lap circles.* *OT* passes through *t*.

From *C* to *M* expansion occurs; and a peculiarity of this type of valve is that from *M* to *R* the steam is compressed.

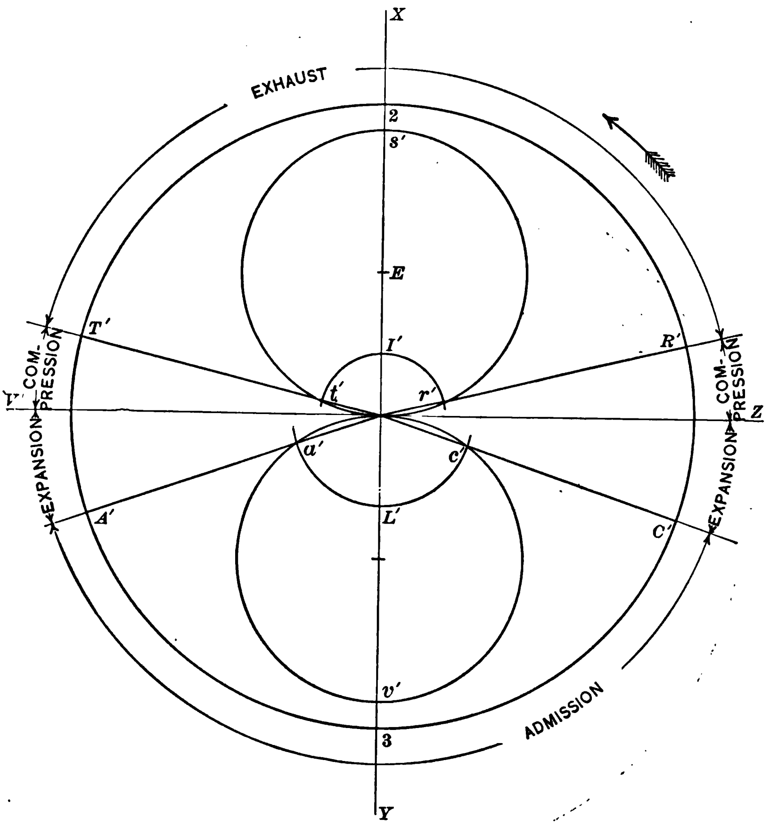



FIG. 27

This is evident because the piston reverses the direction of its motion, and the exhaust does not open until the piston has passed over a portion of its return stroke. In the same way, compression occurs from *T* to *N*, and from *N* to *A* this compressed steam expands.

## Case IV.

Simple D-valve, with both inside and outside laps, with the angular advance sufficient to open the port to steam at the beginning of the stroke.

The diagrams for this case are shown in Figs. 28 and 29,

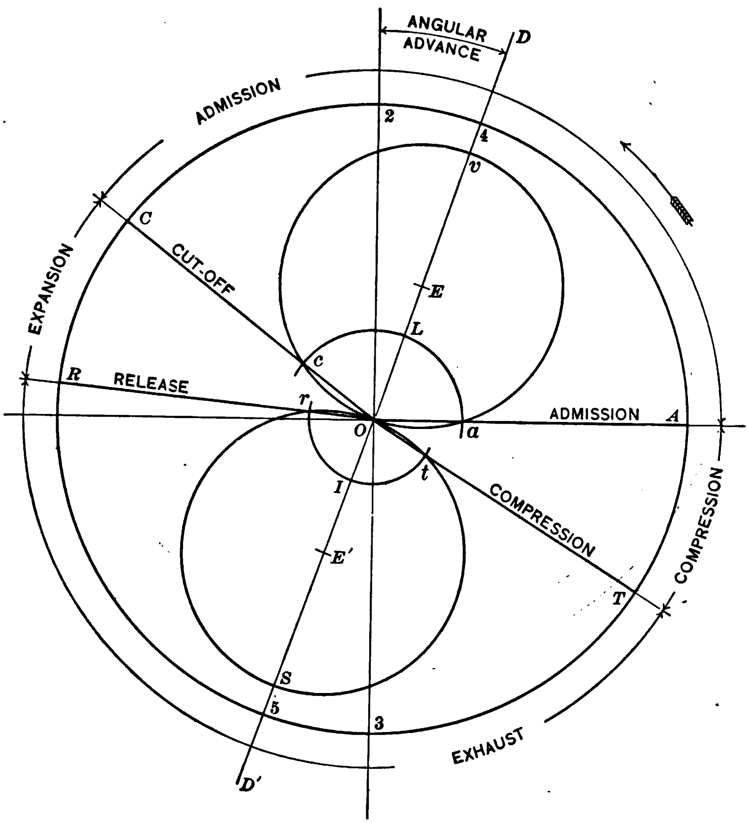



Fig. 28

Fig. 28 being for the head-end. These figures, as in the previous cases, are lettered alike, the sole difference being that

the letters are primed in the crank-end diagram. The notation shown and used thus far will be used throughout the work.

In order to make the valve-circle pass through the intersection of the outside lap-circle and the valve-circle, it is

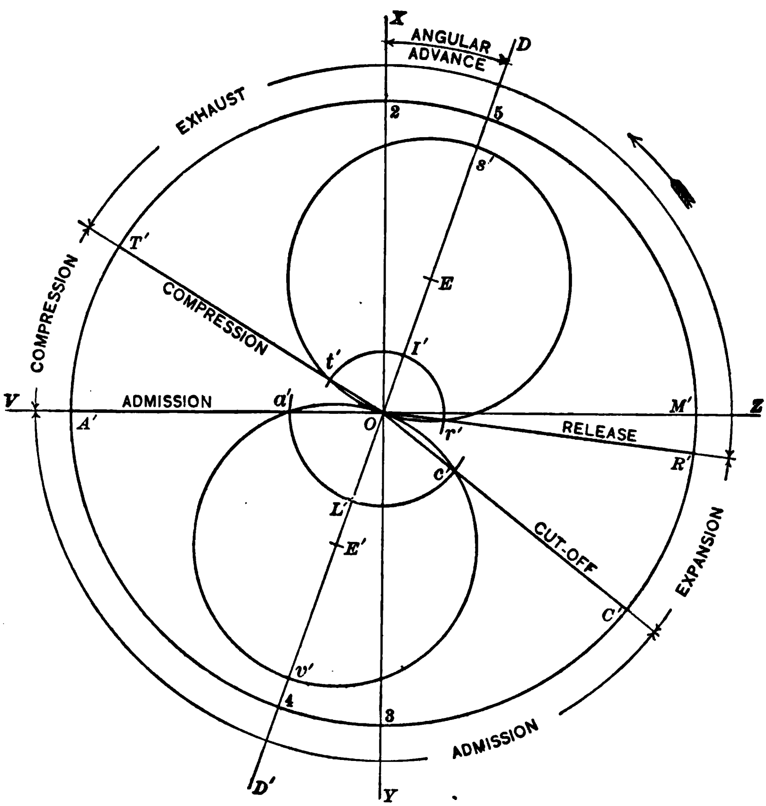



FIG. 29

necessary to incline the line $DD'$ joining $E$ and $E'$, as shown in the cuts. The angle $DOX$ is equal to the angular advance; and it makes no difference whether the eccentric leads or follows the crank—that is, whether the angle between the

crank and eccentric is 90° + the angular advance, or 90° − the angular advance—the line $DD'$ is always inclined as shown, with the upper end, $D$, brought over toward the cylinder. The angular advance will take care of itself when it comes to the valve-setting. Nor does it make any difference whether the cylinder is to the right or left of the crank-shaft as it is located in the diagram. The diagrams are for the head-end and crank-end, which disposes of that question. Neither does it make any difference whether the engine runs over or under. This is one of the beauties of the diagram—that it fits any and all circumstances.

The dimensions of the valve being taken the same as in the preceding case, it will be seen at once that the diagram shows the effect of angular advance to be, as stated in Chapter I, that of making all the events of the stroke earlier.

While the crank is traveling from 5 to 4 the valve moves in one direction; and while the crank travels from 4 to 5 the motion of the valve is reversed. This will help to explain why the facts relating to exhaust and admission are not read from the same valve circles. For, referring to Figs. 23 to 28, it will be seen that when the valve leaves its central position and moves to the left, it uncovers the head-end port to steam, and continues to leave it open until it is again central on its travel to the right. That is, the steam-opening is affected by the last half of the left-hand travel of the valve and the first half of the right-hand travel. Then, in all the valve diagrams, $Ov$ represents the last half of the travel in one direction, and $vO$ represents the first half of the travel in the other direction. The exhaust on the head-end, Figs. 3 to 8, is affected in the opposite way, that is, by the first half of the left-hand travel and the last half of the right-hand travel, and is therefore read from the other valve-circle, where $Os$ is travel in one direction, and $sO$ is travel in the other.

## CASE V.

Simple D-valve, both inside and outside laps, and having the angular advance sufficient to open the port before the beginning of the stroke.

This case, which is shown in Figs. 30 and 31, simply

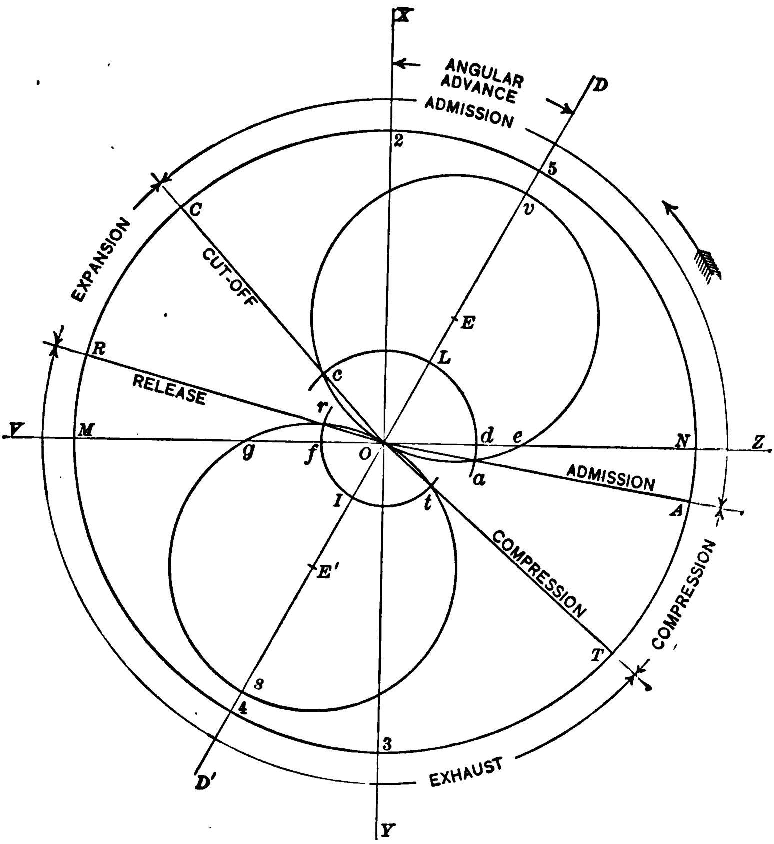



FIG. 30

amounts to moving the eccentric a little farther ahead. Admission occurs when the crank is at *A*. The amount *de*,

which the port is opened when the piston begins its stroke, is called the "steam-lead," or simply "lead." The angle *ZOA* is called the "angle of lead," or "lead angle.'

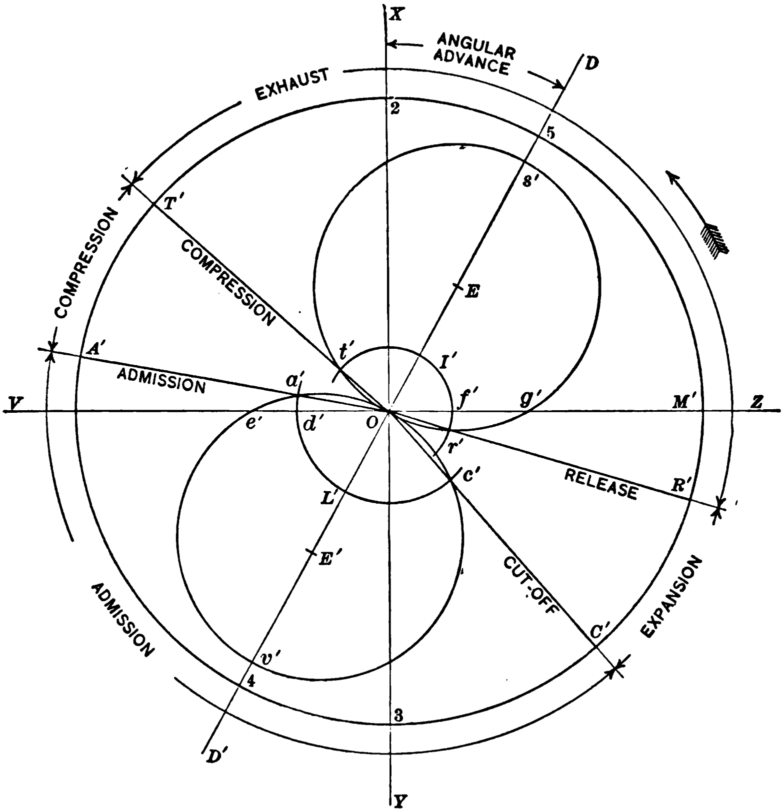



FIG. 31

Similarly, *fg* is the exhaust-lead, and *ROM* is the exhaust-lead angle.

*The "lead" angle is the angle between the crank line at admission and the line of stroke. ZOA is the lead angle.*

This case is the general one. The instructions for drawing it may be summed up as follows:

Draw the two lines *XY* and *VZ* at right angles, intersecting at *O*, which represents the center of the crank-shaft, *VZ* representing the line of stroke. About *O* as a center, and with a radius *OM* equal to the crank, describe the crank-circle whose diameter is *MN*, which is equal to the stroke of the engine. Lay off the angle *XOD* equal to the angular advance. Draw *DOD'*. Lay off *OE* equal to half of the eccentricity, and with *E* as a center and a radius *OE* describe the upper valve-circle. Locate *E'* and describe the lower valve-circle in the same way. (It is neither necessary nor advisable to use the same scale for the valve and crank-circles. The crank-circle is only employed to determine the inclination of the crank to the line of stroke, and need be drawn only on a small scale.) About *O* as a center, and with a radius *OL* equal to the outside lap, describe the outside lap circle. About *O* as a center, and with a radius *OI* equal to the inside lap, describe the inside lap circle.

The diagram is then read in accordance with the rules given.

## EFFECT OF CHANGING VARIOUS DIMENSIONS OF THE VALVE, AS SHOWN BY THE DIAGRAM, WHEN THERE IS ANGULAR ADVANCE.

*Angular Advance.*—The effect of changing this is shown in Figs. 28 and 30, or 29 and 31. If the angular advance is increased, its effect is to make all the events of the stroke earlier. If it is decreased, all the events occur sooner. Refer to Table III.

*Valve Travel.*—If this is increased, cut-off and compression come later and admission and release occur earlier. If it is decreased, cut-off and compression occur earlier and admission and release come later. Refer to Fig. 32 and Table III.

*Outside Lap.*—Increasing this makes admission later and cut-off earlier, prolongs the expansion and compression, decreases the lead, and leaves the exhaust unchanged. Decreasing the outside lap increases the lead, makes cut-off later

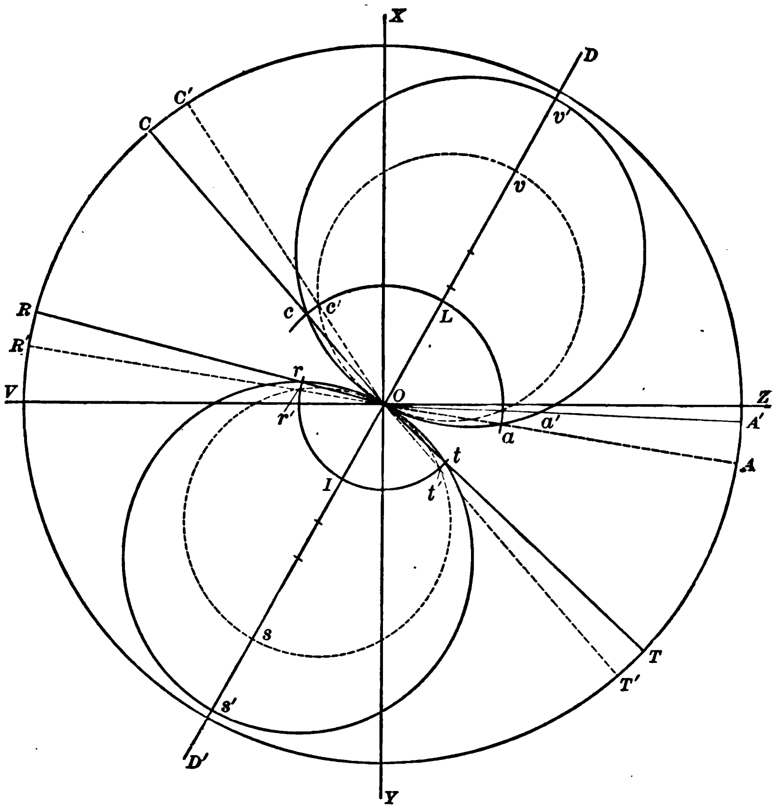



FIG. 32

and admission earlier, shortens expansion and compression, and does not affect the exhaust. Refer to Fig. 33 and Table III.

*Inside Lap.*—Increasing this does not affect lead, admission, or cut-off, but it makes release later and compression earlier,

thus prolonging expansion and compression. Decreasing the inside lap makes release earlier and compression later, thus

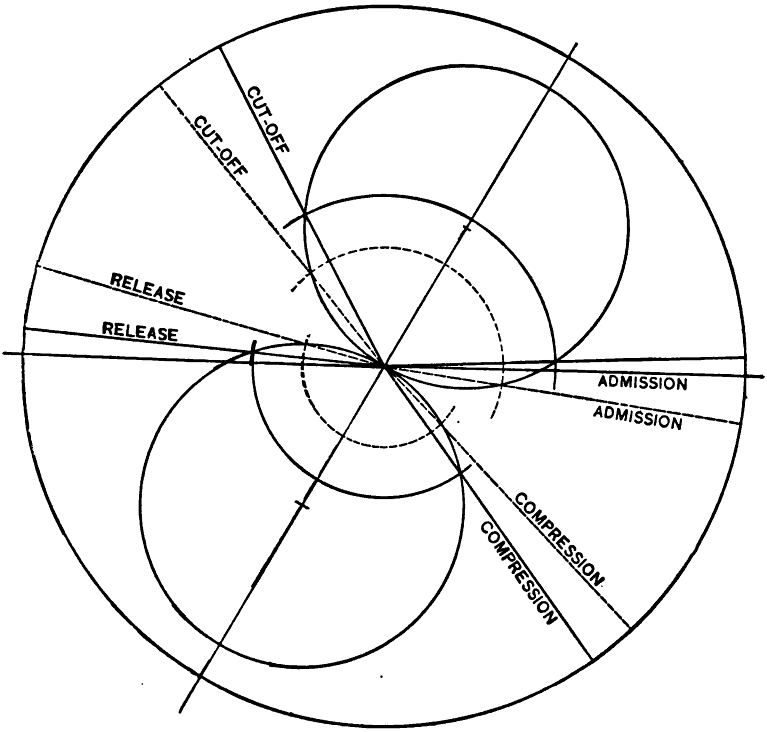



FIG. 33

shortening both compression and expansion. See Fig. 33 and Table III.

## CHAPTER IV.

### DIMENSIONS OF PORTS, STEAM-PIPES, AND BRIDGES.

A STUDY of the valve diagrams will show at once that the dimensions of the valve are influenced very strongly by the point of cut-off. Next in importance to this, as a controlling influence, is the width of the steam-port. This latter is not assumed arbitrarily, but is calculated with a reasonable degree of accuracy.

When steam passes through an opening at a greater speed than 6000 or 8000 feet per minute it is choked, throttled, or "wire-drawn"; that is, its pressure after passing through the opening is less than the pressure urging it through. For this reason the steam must not be forced to pass through the ports at a greater speed than 6000 feet per minute. Now, in order to fill the cylinder up to cut-off with steam at boiler-pressure, it is necessary that the volume of steam admitted shall be equal to the volume swept through by the piston.

The volume of the cylinder up to cut-off is of course equal to the area of the piston multiplied by the distance through which the piston has traveled. If the piston area is one square foot, and the travel up to cut-off is 2 feet, then $1 \times 2 = 2$ cubic feet of steam must be admitted while the piston is moving through those two feet. The amount of steam to be supplied can also be found by multiplying the area of the piston in square feet by the piston speed in feet per minute, and multiplying this result by the time in minutes that the port is open. This may be expressed in a formula as follows:

$$C = A \times S \times T, \quad \dots \dots \quad (1)$$

where $C =$ volume in cubic feet up to cut-off;
$A =$ piston area in square feet;
$S =$ piston speed in feet per minute;
$T =$ time in minutes that port is open.

For example, if the piston area is 1 square foot, the piston speed 400 feet per minute, and the port is open $\frac{1}{200}$ of a minute, then

$$C = 1 \times 400 \times \frac{1}{200} = 2 \text{ cubic feet.}$$

Now, this steam must be admitted through the port, which is open for the same length of time. The volume of steam thus admitted is equal to the velocity of the steam in feet per minute multiplied by the time the port is open, and this result then multiplied by the area of the port in square feet. Expressed in a formula this is

$$C = V \times P \times T, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $C =$ volume admitted in cubic feet;
$V =$ velocity of steam in feet per minute;
$T =$ time in minutes that port is open.

For example, if the velocity of the entering steam is 6000 feet per minute, the port area $\frac{1}{15}$ of a square foot, and the port is open $\frac{1}{200}$ of a minute, then

$$C = 6000 \times \frac{1}{15} \times \frac{1}{200} = 2 \text{ cubic feet.}$$

The results given by formulas (1) and (2) must be equal, of course, so that

$$A \times S \times T = V \times P \times T,$$

or, as $T$ is on both sides, it may be dropped, giving

$$A \times S = V \times P. \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In designing a valve, the area of the piston is known, and the velocity of the steam is assumed in accordance with the statement made above. The piston speed is found by multiplying the length of the stroke by twice the number of revolutions per minute; because the piston makes two strokes for every revolution. That is,

$$S = 2NL, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

where $N =$ number of revolutions per minute;
$L =$ length of stroke in feet.

Now, from formula (3),

$$P = \frac{A \times S}{V},$$

or

$$\text{port area} = \frac{\text{piston area} \times \text{piston speed}}{\text{velocity of steam}}.$$

This may be expressed in the following:

RULE.

*To Find the Port Area for a Given Engine: Multiply the piston area in square feet by the piston speed in feet per minute, and divide the product by the velocity of the steam, in feet per minute. The result will be the area of the port in square feet.*

For example, take an engine 18 × 24—that is, one in which the cylinder is 18 inches in diameter and 24 inches stroke,—running at 200 revolutions per minute, and assume the velocity of the steam to be 6000 feet per minute. To find the port area required. The piston speed is, according to formula (4),

$$2 \times \frac{24}{12} \times 100 = 400 \text{ feet.}$$

The 24 is divided by 12 to reduce the stroke to feet. The area of an 18-inch circle is 254.47 square inches, from Table V at the end of the book, or $254.47 \div 144 = 1.766$ square feet.

Now, applying the rule, $1.766 \times 400 = 706.4$, and $706.4 \div 6000 = .1174$ square feet, or 16.91 square inches.

The work of this is renderedve ry light by using Table IV, which contains the values of

$$\frac{\text{piston speed}}{\text{velocity of steam}}$$

for various velocities of steam, and for piston speeds from 100 to 1200 feet per minute, advancing by 25. It is only necessary to multiply the figure given in the table by the piston area to get the port area. Take the example just worked. Under 6000, in the column headed "Port Area, Piston Area as Unity," and opposite 400 feet piston speed, will be found .067. Multiplying the area of piston, 254.47 square inches, by .067, gives 17.049 square inches. The difference between this result and the one obtained by the longer method is very slight, only .13 of a square inch, and is due to the use of decimals.

The length of the port should be made, as nearly as possible, equal to the diameter of the cylinder, and the width is of course found by dividing the area found as above by the length. If the length of the port is nine tenths of the diameter of the cylinder, the width may be found by multiplying the diameter of the piston by the figure given in Table IV. For example, in the engine just considered, if the port length is .9 of the cylinder diameter, the port width is found by mutiplying 18 (the diameter of the cylinder) by .058, which is the figure opposite the piston speed, 400, and under "6000, Width of Port." This gives $18 \times .058 = 1.024$ inches as the width of the port.

If the velocity of the steam were assumed as 4000, the multiplier would be .086. If the velocity were 8000 and the piston speed 300, the multiplier would be .033.

After the steam has been expanded in the cylinder it is at a lower pressure than when it was admitted, and will travel

at a lower rate of speed. The exhaust-ports should therefore be made with a greater area than the steam-ports. In order to allow for this it is usual to assume that the velocity at entrance is 6000 feet per minute, and at exhaust is 4000 feet per minute. These velocities are exceeded in certain cases, such as locomotives, and the table has been extended to cover these excesses; but throughout this work the figures just mentioned will be employed.

The area of the exhaust-ports may be found by applying the rule given, or the table may be employed.

Of course when the same port is used for admission and exhaust, as in the case of the common slide-valve, the port width must be made that corresponding to the lower velocity, in order that the exhaust may be free and unimpeded.

The diameter of the steam- and exhaust-pipes for a given engine may be determined by the following rule:

*To Find Diameter of Steam- and Exhaust-pipes:* Multiply the diameter of the piston by the square root of the port area; the latter being expressed as a fraction of the piston area.

This rule is obtained as follows:

The piston area is taken as unity. As explained before, the port area is a certain fraction of the piston area. Represent this fraction by $F$. The areas of two circles are to each other as the squares of their diameters, so that

$$1 : F :: D^2 : d^2,$$

where $1 =$ piston area;

$F =$ port area, as a fraction of piston area, given in column of table headed "Port Area";

$D =$ diameter of piston;

$d =$ diameter of pipe.

This formula gives

$$d = D\sqrt{F}.$$

In order to save the trouble of the extraction of the square root, a column in Table IV, headed "Diameter of Steam-Pipe," has been provided, which contains the values of $\sqrt{F}$ corresponding to different piston speeds. This column is used in the same manner as the others. For instance, for the engine just discussed, 18 inches diameter and 400 feet piston-speed, the steam-pipe diameter is $18 \times .258 = 4.82$ inches; the figure .258 being under "6000, Diameter of Steam-Pipe" and opposite 400. The exhaust-pipe diameter is found from he table by assuming the lower velocity. In this case it would be $18 \times .316 = 5.688$ inches.

Suppose an engine designed for a slide-valve having a port width $E$, Fig. 34, equal to that required for free exhaust. If the port is opened its full width, the valve assumes the position shown in the figure when at the end of its travel. It is only necessary, however, to open the steam-port sufficiently to allow a free admission; that is, the valve assumes the position shown in Fig. 35, $A$ being the width required for unchoked admission. By designing a valve to open the port only the smaller distance, the travel can be decreased by the difference between $E$ and $A$. This will not interfere with a free opening for exhaust, as this event is controlled by the inner edge of the valve; and the lap there can be so modified as to give the desired opening. As shown in Fig. 35, the exhaust is fully open.

When the valve has reached its extreme position, one port is open to its greatest extent to admit steam, and the other one is opened as fully as possible to exhaust. The amount that the port is open to steam in this position is called the "maximum port-opening."

*The maximum port-opening* is made equal to the width of port necessary for free admission of steam.

*The maximum opening for exhaust* is equal to the width of the port.

When the travel of the valve is so great that the valve

assumes the position shown in Fig. 36 when in its extreme position, the amount $x$ that it runs over the edge is called the *overtravel.*

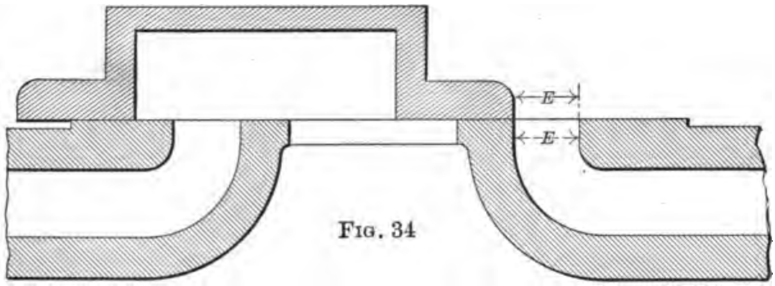



FIG. 34

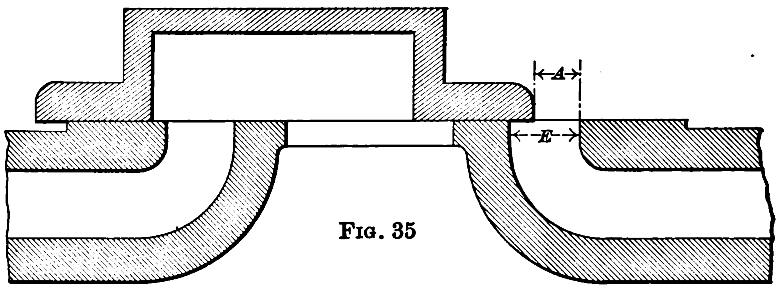



FIG. 35

In order to fix these principles, as well as to illustrate the use of the tables again, another example will be worked out.

GIVEN. An engine 20 × 30, running at 150 revolutions. Length of ports, 19 inches.

REQUIRED. (1) Width of port. (2) Diameter of steam-pipe. (3) Diameter of exhaust-pipe. (4) Maximum port-opening.

SOLUTION. (1) Piston speed $\frac{150 \times 20 \times 2}{12} = 500$ feet per minute. See formula (4). In Table I, under "4000, Area of Port" and opposite 500, is .125. The area of a 20-inch circle is 314.16 square inches. $314.16 \times .125 = 39.27$ square

inches. $39.27 \div 19 = 2.668$. Nearest sixteenth, $2\frac{11}{16}$. Width of port $= 2\frac{11}{16}$.

(2) In Table I, under "6000, Diameter of Steam-Pipe" and opposite 500, is .288. Then the diameter of steam-pipe should be $20 \times .288 = 5.76$ or $5\frac{3}{4}$ inches.

(3) In Table I, under "4000, Diameter of Steam-Pipe"

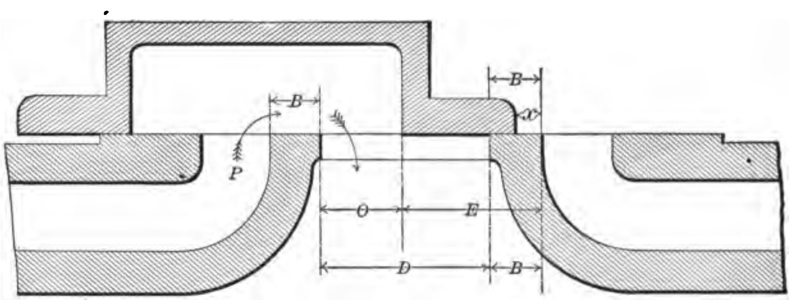



**Fig. 36**

and opposite 500, is .353. Then the diameter of the exhaust-pipe should be $20 \times .353 = 7.06$ or $7\frac{1}{16}$ inches.

(4) The velocity of the steam at exhaust being 4000, while that at exhaust is 6000, it follows that the port need only be opened $\frac{4000}{6000}$ of its width to secure free admission. This is equal to $\frac{2}{3}$. The maximum port-opening is therefore $\frac{2}{3} \times 2\frac{11}{16} = 1.79$ inches. It is therefore made $1\frac{13}{16}$ inches.

The same result would be obtained by using the table.

That part of the cylinder-casting which divides the steam- and exhaust-ports is called the "bridge." *B*, Fig. 36, is the width of the bridge. This width must be sufficient so that if the valve has overtravel—that is, if the end of the valve runs beyond the edge of the port, as shown in the figure—the steam- and exhaust-ports will not be in communication. If the greatest opening of the port does not exceed the width of the port, it is sufficient to make the bridge width equal to the thickness of the cylinder-casting, which makes it comparatively easy to obtain a good casting; but if the valve has

overtravel greater than the thickness of the cylinder-walls, it is necessary to increase the thickness of the bridge; which may be summed up in the following rule:

*The Width of the Bridge* is made equal to the thickness of the cylinder-wall when the valve has either no overtravel or an overtravel less than the thickness of the cylinder-walls. When the overtravel exceeds this amount, the bridge thickness is found by adding $\frac{1}{4}$ inch to the port-opening and subtracting the width of the port from the sum. The remainder is the thickness of the bridge.

Fig. 36 shows the valve in its extreme left-hand position. The exhaust is then passing through the port $P$ and the opening $O$. These two openings must be equal in order that the exhaust shall not be cramped; for it was determined that the width $P$ is necessary for free exhaust from the cylinder, and the same width is evidently necessary at the exhaust, in order not to increase the velocity of steam at exhaust to such an extent as to increase the back pressure. The valve shown in the figure has no inside lap; and as the valve has moved a distance $E$, equal to one half the valve travel from its middle position, the distance $D$, which is the width of the exhaust-port, must be equal to $E + O - B$; but $O$ is equal to $P$, so that

$$D = E + P - B.$$

If the valve has inside lap, the width of the exhaust-port must be increased by the amount of the inside lap.

*To Determine the Width of the Exhaust-port:* 1. *When the valve has no inside lap, add together the width of the steam-port and one half the valve travel. From the sum subtract the width of the bridge. The remainder is the width of the exhaust-port.* $D = E + P - B$.

2. *When the valve has inside lap, add together the width of the steam-port and the eccentricity or half-travel of the valve. From the sum thus obtained subtract the width of the bridge,*

*and to the remainder add the amount of the inside lap. The sum is the width of the exhaust-port.* $D = E + P - B - I$.

3. *When the inside lap is negative—that is, when the valve allows the port to be open to exhaust when the valve is in its middle position—add together the half-travel or eccentricity and the width of the steam-port. Then add together the amount of the negative lap or inside clearance and the thickness of the bridge. Subtract this latter sum from the first, and the remainder is the width of the exhaust-port.* $D = E + P - B - I$.

# CHAPTER V.

## VALVE DIAGRAMS—GENERAL PROBLEMS OF DESIGN.

THERE are several different problems which may present themselves for solution, circumstances beyond the designer's control having fixed certain dimensions. But the following problems will cover the probable cases very completely.

### PROBLEM I.

GIVEN. Eccentricity, point of cut-off, angle of advance, and point of compression.

REQUIRED. Lap, exhaust-lap, lead, exhaust-lead, and greatest possible openings of port to both admission and exhaust.

SOLUTION. The solution of this problem is shown in Fig. 37. Draw $XY$ and $VZ$ at right angles, intersecting at $O$. Draw $DOD'$, making the angle $DOX$ equal to the given angle of advance. Find $CO$, the crank position corresponding to the given point of cut-off by either of the two methods previously described. It is best to make the determinations of crank positions on a separate sheet, and transfer them to the sheet on which the solution is being made. Then lay off $ov$ equal to the eccentricity or half-travel. Find $E$, the middle point of $ov$, and about $E$ as a center, and with a radius equal to $EO$, describe the upper valve circle. In a similar way find $E'$ and draw the lower valve circle.

Now the upper valve circle intersects $oc$ at $c$. Then, according to Chapter III, $oc$ must be the outside lap. Then

draw in the lap circle about $O$ as a center and with a radius equal to $oc$. This will cut $VZ$ at $d$, and the distance $de$ from $d$ to $e$, the intersection of the valve circle with $VZ$, is the lead. The lap circle cuts the valve circle at $a$, and therefore

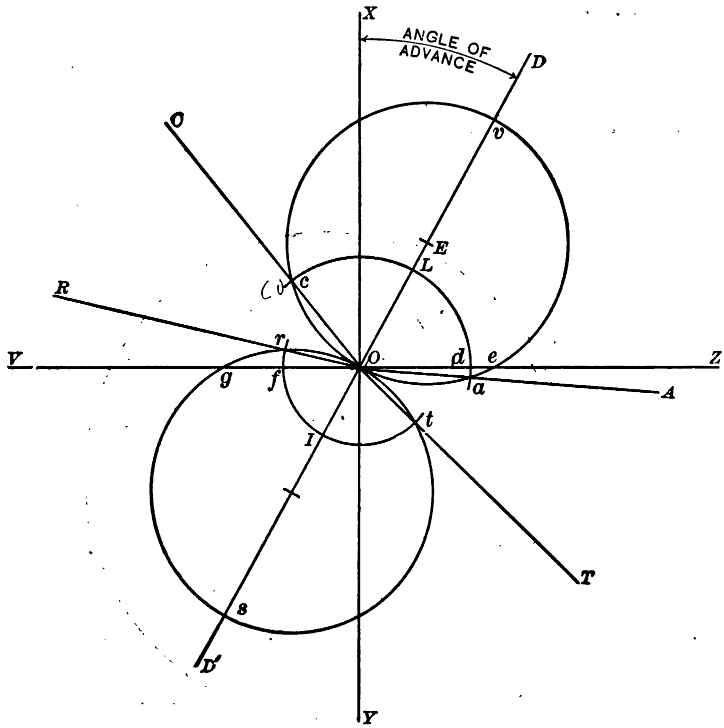



FIG. 37

$OA$, drawn through the point $a$ is the crank-position at admission. The angle $ZOA$ is the lead angle.

Now draw in $OT$, the crank-position corresponding to the given point of compression. This intersects the lower valve circle at $t$, and $Ot$ is therefore the amount of the inside lap. The inside lap-circle is then drawn in about $O$ as a center and with $Ot$ as a radius. Its intersection with the

valve-circle at *r* determines *OR*, the crank-line at release. The exhaust lead is *fg*, and the exhaust lead angle is *VOR*.

## PROBLEM II.

GIVEN. The lap, point of cut-off, and lead.

REQUIRED. The valve travel and angle of advance.

SOLUTION. Fig. 38 shows one solution of this problem. The lines *XY* and *VZ* are drawn at right angles, intersecting

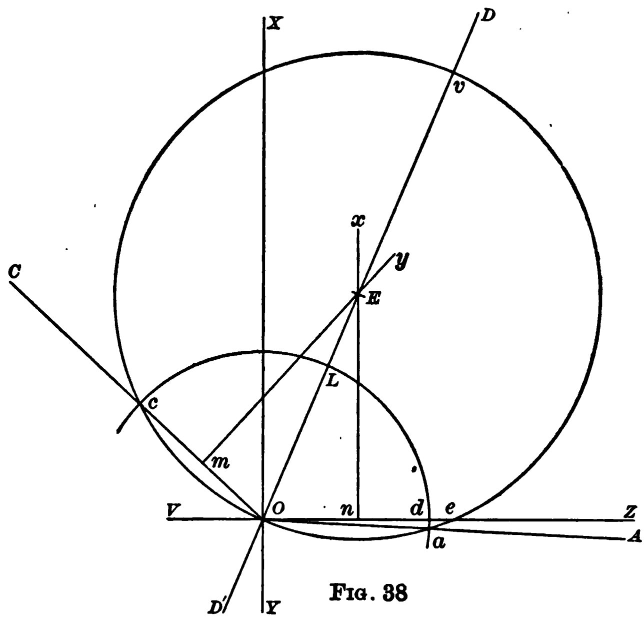



FIG. 38

at *O*. The crank-position at cut-off is *OC*, as given. Then the lap-circle is drawn in about *O* as a center and with a radius equal to the given lap. Its intersection *c* with *OC* is a point through which the valve-circle must pass. *O* is another point on the valve-circle. Another point is *e*, found by laying off *de* equal to the given lead. These three points

serve to determine the center of the valve-circle, as follows: Find *m*, the middle point of *OC*, and draw *my* perpendicular to *Oc*. Find *n*, the middle point of *Oe*, and draw *nx* perpendicular to *Oe*. Then *nx* and *my* intersect at *E*, which is the center required. Then draw in the valve-circle about *E* as a center and with *EO* as a radius. Draw *DEO*, and its intersection *v* with the valve-circle determines *Ov*, which is the eccentricity. The valve-travel is twice the eccentricity.

Another solution is shown in Fig. 39. Having found the points *c* and *e*, draw *ev* perpendicular to *VZ*. Draw *cv* per-.

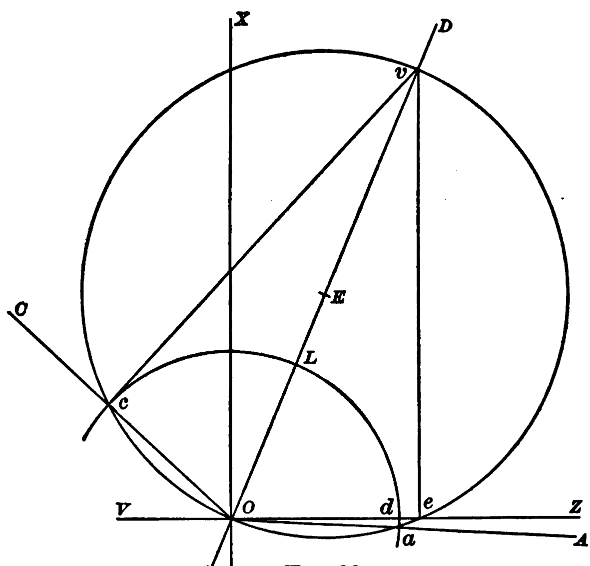



FIG. 39

pendicular to *OC*, intersecting *ev* in *v*. Then draw *DvO*. Then is *vO* equal to the valve travel. If the middle point, *E*, of *vO* is located, and a circle of radius *EO* drawn about *E* as a center, it will pass through *c* and *e* as well as *O*, showing the construction to be correct.

## PROBLEM III.

GIVEN. Cut-off, angle of lead, width of port, and over-travel.

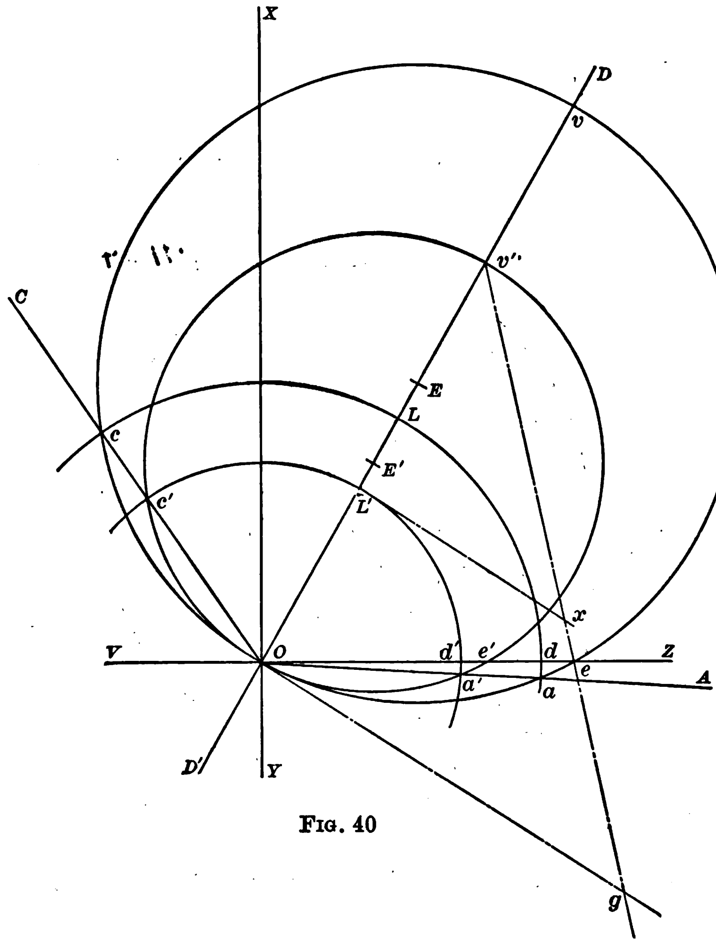



FIG. 40

REQUIRED. Eccentricity, lap, lead, and angle of advance.
SOLUTION. Draw $XY$ and $VZ$, Fig. 40, at right angles to

each other. Draw $OA$, the crank position corresponding to the given lead angle; that is, make $ZOA$ equal to the given angle. Draw $OC$, the crank-line at cut-off, as given. Now, the center of the valve-circle must lie somewhere on the line bisecting the angle formed by these two crank-lines. Therefore draw $DOD'$, bisecting the angle $COA$, and the angle $XOD$ is the angular advance.

Take any convenient radius, as $E'O$, and describe a *trial* valve-circle, intersecting $OD$ at $v'$. Then $Oc'$ is the corresponding lap, and by drawing in this *trial* lap-circle it will be found that the greatest possible opening of the port is $L'v'$, with this lap and valve-travel. The greatest possible opening of the port is evidently equal to the width of the port plus the overtravel, and as both of these are given in the problem, the maximum port-opening is known. It is hardly probable, though possible, that the distance $L'v'$ will equal the given amount on the first trial. That being the case, proceed by drawing $v'g$ in any convenient direction, and take $v'x$ equal to the required greatest port-opening—that is, equal to the width of the port plus the overtravel. Then join $L'$ and $x$, and from $O$ draw $Og$ parallel to $L'x$, until it cuts $v'g$ in $g$. Then is $v'g$ equal to the required eccentricity. Next lay off $Ov$ on $OD$, equal to $v'g$. Bisect $Ov$ in $E$, and about $E$ as a center and with a radius $OE$ draw in the valve-circle which will cut $OC$ at $c$, thus determining the lap, $Oc$; and by drawing the lap-circle the lead $de$ is determined.

The maximum port-opening with this travel and lap is evidently $vL$; and if the construction is accurate, $vL$ will be found to be equal to $v'x$.

### PROBLEM IV.

GIVEN. Point of cut-off, lead, and greatest possible port-opening.

REQUIRED. Lap, valve-travel, and angle of advance.

SOLUTION. This is shown in Fig. 41. Draw $XY$ and $VZ$ at right angles, intersecting at $O$. Draw $OC$, the crank-line at cut-off, as given. Then produce $OC$ to $m$, making $Om$ equal to the given lead. This should be drawn on a large scale, as the lead is usually small, seldom exceeding $\frac{3}{16}$ of an

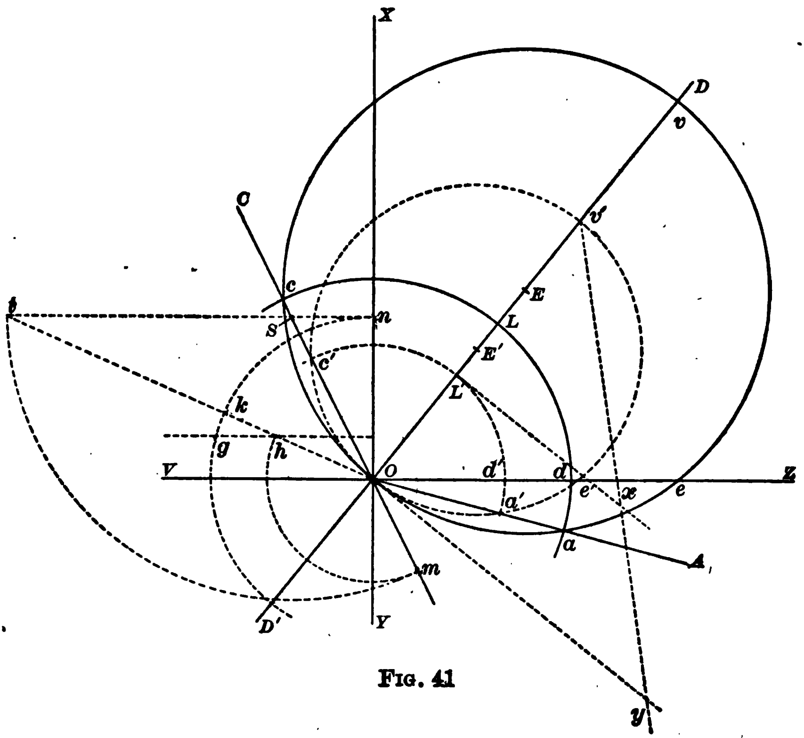



FIG. 41

inch. A scale of from two to four times full size should be employed.

Next take $ms$ on $mC$, equal to the given maximum port-opening. Through $s$ draw $nst$ parallel to the line of stroke $VZ$, making $st$ equal to $sm$ as shown by the arc $mt$. Then join $O$ and $t$, and on $Ot$ take $Oh$ equal to $Om$ as shown by the

arc *mh*. Next about *O* as a center, and with a radius equal to *On*, describe an indefinite arc. Through *h* draw *gh* parallel to *VZ*. Then *gh* will cut the arc just drawn in *g*. The arc cuts *Ot* in *k*, and this arc *kn* is next picked up in the compass and laid off from *g* to *D′*. Then draw *D′O* and produce it indefinitely, as to *D*. Then is the center of the valve circle on *D′OD*, and *DOX* is the angle of advance.

The problem now reduces to:

GIVEN. Cut-off, lead, angle of advance, and greatest port-opening.

REQUIRED. Eccentricity.

This is the same as Problem III, and the solution is made in accordance with the directions given for that. The drawing, Fig. 41, covers this solution, and is lettered the same as Fig. 40 to enable the construction to be followed throughout, if desired.

The proof of these solutions is in one or two cases rather mathematical, and is therefore omitted, as it is entirely unnecessary to understand this proof in using the diagrams.

In order to fix these principles, it is not sufficient to rest content with merely reading the foregoing text, but it is necessary to solve numerous examples under each problem.

Other problems will probably suggest themselves to the mind, but they can either be reduced to the preceding ones, or can be solved by the application of the general principles of the diagram as outlined in Chapter III.

## CHAPTER VI.

### ROCKERS AND BELL-CRANKS.

IT has been said that the motion of the eccentric is very often imparted to the valve through a rocker which joins the end of the eccentric-rod to the end of the valve-stem. This is a necessary appliance when the path of the valve is not in the same straight line, that is, when the line of valve-travel is parallel to the line of stroke, but is above or below it, or on one side of it.

Figs. 42 and 43 show two forms of this rocker, the principal difference between them being that with the plain rocker, Fig. 42, where the pivot is at the bottom, the eccen-

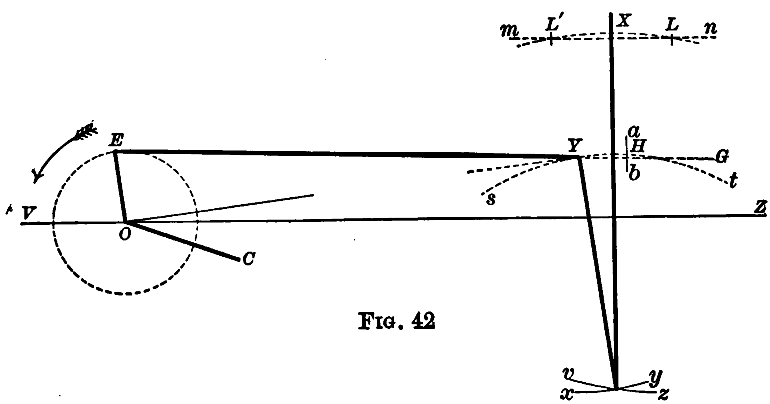



FIG. 42

tric-rod at the top, and the valve-stem attached somewhere between, the valve motion is the same in direction as that of the eccentric, and therefore the eccentric precedes the crank,

as shown in the drawing; while with the bell-crank shown in Fig. 43, with the pivot at the angle, the motion of the valve is opposite in direction to that of the eccentric, and therefore the eccentric follows the crank. In both cases the *amount* of motion of the valve depends upon the relative lengths of the

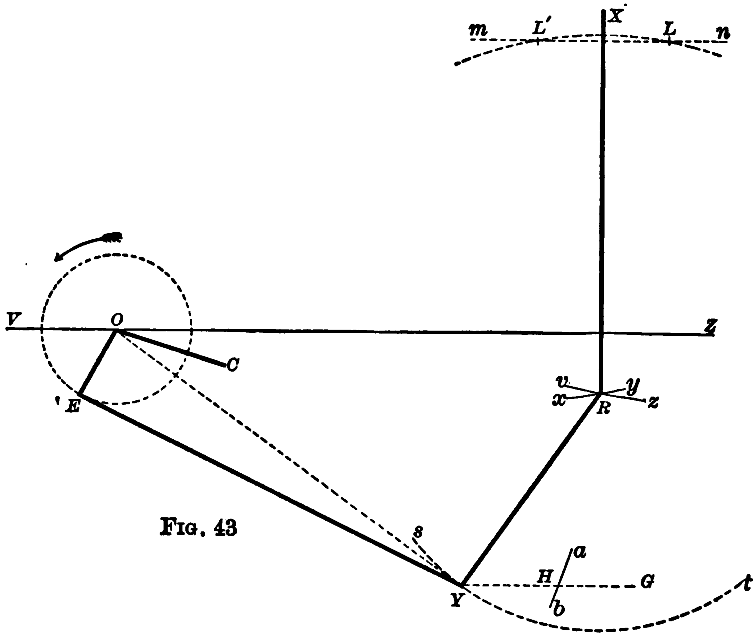



FIG. 43

rocker-arms to which the valve-stem and eccentric-rods are attached.

The method of laying out these rockers will be understood by reference to the figures, which are lettered alike, as the same method is pursued in both cases.

Let *VZ* be the line of stroke, and *O* the center of the shaft. Then take *O* as a center, and with a radius *OE* equal to the eccentricity describe the eccentric-circle, as shown. Let the direction of rotation be as represented by the arrow. Then draw *mn* at the proper distance from *VZ* and parallel to

it, to represent the line of travel of the valve-stem, and let $X$ be the position of the end of the valve-stem when the valve is in its middle position. Lay off $XL$ and $XL'$, each equal to the outside lap of the valve. Then take $L$ and $L'$ as centers, and with radii equal to the length of the longer arm of the rocker describe the two arcs $vz$ and $xy$, intersecting at $R$. Then is $R$ the pivot of the rock-shaft. Then take $R$ as a center, and a radius equal to the other arm of the rocker, and draw the arc $st$, making it of indefinite length. From $O$ draw $OY$ tangent to the arc $st$, determining the point of tangency, $Y$, by drawing $RY$ perpendicular to $OY$, the point of intersection being the required point. Next, from $O$, draw $OE$ perpendicular to $OY$. Then $OE$ represents the eccentric, and by joining $E$ and $Y$ the length $EY$ is determined, which is the length of the eccentric-rod.

Peculiarities of the design may make it an object to make the eccentric-rod of a certain length. This may be done by shifting the pivot of the rocker until the desired length is obtained. If the distance through which the pivot is to be so moved is short, proceed as follows:

Through $Y$ draw a line $YG$ parallel to the line of stroke $VZ$. With $E$ as a center, and with a radius $Ea$ equal to the given length of the eccentric rod, describe an arc $ab$, intersecting $YG$ in $H$. Then move the whole arrangement through the distance $YH$, changing the length of the valve-stem by that amount. The same construction would hold good if the rod were to be shortened instead of lengthened.

With this construction the action of the valve is not made irregular at either admission or cut-off. The angle between the crank and eccentric is changed, but this is a matter which affects the valve-setting alone. The amount of the change may be found by laying off $ZOC$ equal to the angle of advance as found by the valve diagram, and then $OC$ represents the position of the crank with the valve in its middle position, and $COE$ is the angle between the crank and eccentric.

Sometimes the valve-face is not parallel to the line of stroke. This is a combination seldom met, as the lack of parallelism renders it a difficult matter to face off the valve-seat and bore the cylinder. Therefore an extended description of this type will be omitted. There is one special case, however, which it is well to bear in mind, and that is when the center line of the valve-stem is inclined to the line of stroke in such a manner that it would, if produced, pass through the center of the shaft. This changes the angle of advance by just the angle between the two lines—the line of stroke and the center line of the valve-stem. If the change is but slight, it may be neglected. In any case, it is a matter which affects the valve-setting alone.

In both Figs. 42 and 43 the rocker-arms are unequal, the shorter arm being $\frac{3}{8}$ of the larger one, the resultant travel of the valve being the same as if the eccentricity were $\frac{3}{8}$ of $OE$. In designing and laying out the valve, it is treated as if it were actuated by the larger eccentric. That is to say, no allowance is made in construction until it comes time to lay out the eccentric, when that is made with, in this case, $\frac{3}{8}$ of the throw which would be given it if it were either connected direct or with an equal-armed rocker.

The valve diagrams given in Chapters III and V show that if the cut-off is equalized on the two ends—that is, if made to occur at the same point on each stroke—by making the outside laps unequal, the leads will be unequal; and if the leads are kept equal, the cut-offs will be unequal. It is possible, however, by the use of a bell-crank designed in a certain way, to equalize the cut-offs and have the leads equal or nearly so. This bell-crank is designed as follows: Determine the outside lap, angle of advance, and eccentricity necessary to secure the required cut-off and lead on *one end* of the valve. It makes no difference which end of the cylinder is selected, as the difference between the laps would be compensated for by the difference between the resulting bell-cranks. Then pro-

ceed to construct the diagram shown in Fig. 44, which should be drawn on as large a scale as possible in order to render the work accurate.

Draw $VZ$ to represent the line of stroke, and let $O$ be the center of the crank-shaft. About $O$ as a center, and with

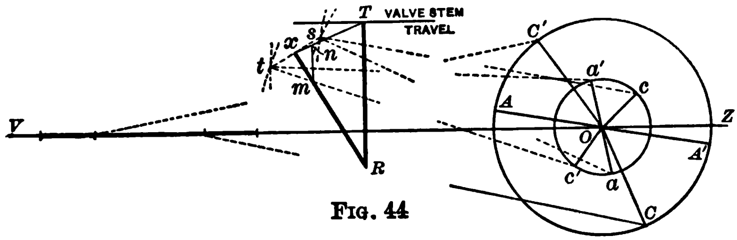



FIG. 44

a radius equal to the length of the crank, describe the crank-circle. Then describe the eccentric-circle about $O$ as a center and with a radius $Oc$ equal to the eccentricity. Next find the crank positions corresponding to the given point of cut-off on the forward and return strokes respectively. These are $OC$ and $OC'$, and may be found by either of the two methods given in Chapter II. The first method is the one adopted here, because the method of Deprez, requiring the use of the auxiliary crank-circle, would complicate the drawing too much at the right-hand end. Next draw in $OA$ and $OA'$, the crank-positions at admission on the forward and return strokes respectively. These positions are determined by taking the crank-position for admission from the end of the cylinder for which the lap, etc., were determined, and making the other lead angle the same. For instance, if $OA$ was found, then the angle $ZOA'$ is made equal to $VOA$. The next step is to locate the eccentric-positions corresponding to these four crank-positions. This is done by laying off from each crank-position the angle between the crank and eccentric. That is, $COc$, $C'Oc'AOa$, and $A'Oa'$ are each equal to the angle between the crank and eccentric.

Now, it must be remembered that when admission takes

place, the valve is just opening the port; and when cut-off occurs, the valve is just closing the port. Consequently the valve is in the same position at admission and cut-off, but the motions are in opposite directions. That is to say, with the eccentric-rod end at either $a$ or $c$ the valve must be in the same position. Then with $a$ and $c$ as centers, and with radii equal to the length of the eccentric-rod, describe arcs intersecting at $S$. Then take the same radius, and with $a'$ and $c'$ as centers describe two more arcs which intersect at $t$. Join $s$ and $t$, and find the middle point $x$. At $x$ draw $xR$, perpendicular to $ts$. Then if a line be drawn perpendicular to the line of stroke, as, for example, $RT$, it will determine the angle $xRT$ between the two arms of the bell-crank. The lengths of these two arms must be such as to give the required movement of the valve, and are determined as follows.

On $xR$ lay off $xm$ equal to $xs$, and from $m$ draw the line $mn$ perpendicular to the line of travel of the valve-stem, and make $mn$ equal to the lap of the valve. Draw $xT$ from $x$ through $n$ until it meets the line of travel of the valve-stem at $T$; and from $T$ draw $TR$, also perpendicular to the line of valve-stem travel until it meets $xR$ at $R$. Then will $TR$ and $Rx$ be the lengths required for the two arms of the bell-cranks, while $R$ will be the point of suspension. The valve-stem is fastened to $T$, and the eccentric-rod to $x$. The general appearance is shown by the heavy lines.

In a similar manner the exhaust and compression can be equalized, obtaining points similar to $t$ and $s$; and if these points should happen to lie in such positions that an arc passing through them would also pass through $s$ and $t$, it would be possible to design a bell-crank which would equalize all the events of the stroke. As an ordinary thing this coincidence is impossible, so it may be taken as an accepted fact that with a plain D valve it is impossible to equalize all the events of the stroke. This is one of the greatest drawbacks to this type of valve.

## CHAPTER VII.

### DESIGN OF A PLAIN D-VALVE.

IF the preceding chapters have been thoroughly mastered, the student is in possession of sufficient information to design a plain slide-valve. In order to fix clearly the various steps and their order, a numerical example will be worked out completely.

Let the engine be $24 \times 24$, running at 135 revolutions per minute. Cut-off to be at $\frac{7}{8}$ stroke, compression at $\frac{9}{10}$ stroke, and the lead $\frac{1}{8}$ of an inch. The connecting-rod is five times the length of the crank. Length of port 22 inches.

### ORDER IN WHICH THE VARIOUS DIMENSIONS ARE TO BE DETERMINED.

1. *Piston Speed.*—This is $\frac{135 \times 24 \times 2}{12} = 540$ feet per minute.

2. *Area of Steam-port.*—This is determined by reference to Table III, assuming that the velocity of steam at exhaust is 4000 feet per minute. Under "4000, Area of Port" and opposite 550, which is the nearest to the given piston speed, is .138. The piston being 24 inches in diameter, its area is 452.39 square inches, and the port area is therefore

$$452.39 \times .138 = 62.430 \text{ square inches.}$$

3. *Width of Steam-port.*—This is the area just found divided by the length given, or

$$62.430 \div 22 = 2.837 \text{ inches.}$$

The nearest sixteenth is $2\frac{13}{16}$, from Table II, which is therefore the width required.

4. *Maximum Port-opening.* — This will be the amount which would allow the entering steam to have a velocity of 6000 feet per minute. It may be found by taking the number from Table IV, multiplying it by the piston area, and then dividing it by the port length; but it is easier to multiply the result given in (3) by $\frac{2}{3}$, as 4000 is $\frac{2}{3}$ of 6000. This gives

$$2.837 \times \tfrac{2}{3} = 1.891 \text{ inches},$$

and, from Table II, the maximum port-opening is made $1\frac{7}{8}$ inches.

5. *Outside Lap, Angle of Advance, Valve-Travel.*—These must be such as to produce the required point of cut-off with the given amount of lead, and the port-opening found in (4). This amounts to an example in Problem IV, given in Chapter V.

First determine the crank-position corresponding to the given point of cut-off on either end—the head-end, for example. This is done, as shown in Fig. 45, by the method given in Chapter II. For this purpose a sheet of paper is required which will allow the drawing to be made on a large enough scale to measure accurately. The crank circle being 24 inches in diameter, a sheet of paper fourteen inches square will be large enough for a half-size drawing. Then draw the two lines $XY$ and $VZ$ at right angles, locating $O$, which will represent the center of the shaft. Then locate $O'$, the center of the *auxiliary* crank-circle. The connecting-rod being five times as long as the crank, $OO'$ will be $\frac{1}{20}$ of the crank, as shown in Table I. That is, $OO'$ will be

$$\frac{12}{20} = \frac{3}{5} = .6 \text{ of an inch},$$

and on the half-size drawing it will be .3 of an inch. Then

draw in the two circles shown, the dotted circle being the auxiliary crank-circle. Lay off *vn* equal to $\frac{7}{8}$ of *vz*, as the cut-off is to be at $\frac{7}{8}$ stroke. Draw *nc* perpendicular to *vz*, cutting the auxiliary circle at *c*, and join *O* and *c*, producing

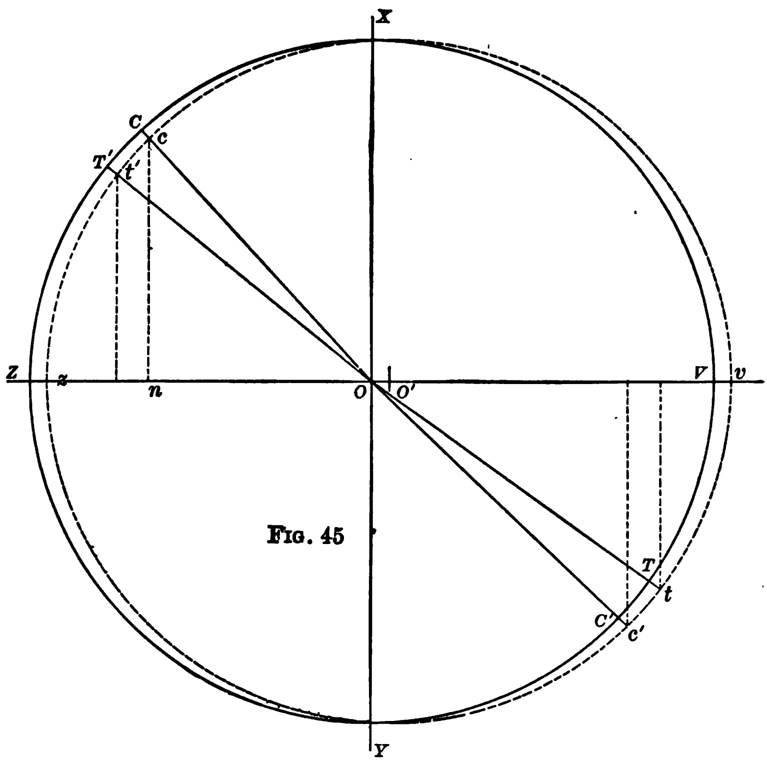



FIG. 45

it until it meets the crank circle at *C*. Then *OC* is the crank position at cut-off on the head end. (Refer to Chapter II for a complete explanation of this method.)

Now, having located the crank-position at cut-off, take another sheet of paper, and proceed as in Problem IV. Draw *XY* and *VZ* at right angles, Fig. 46. Their intersection represents the center of the shaft. Draw *OC*, the crank-

position at cut-off, as found in Fig. 1. Produce *OC* to *m*, making *Om* equal to the given lead, $\frac{1}{8}$ of an inch. This drawing should be made at least twice full size, in which case a sheet of paper 12 inches square will be large enough. Next, on *mC* take *ms* equal to $1\frac{1}{8}$ inches, the given maximum port-

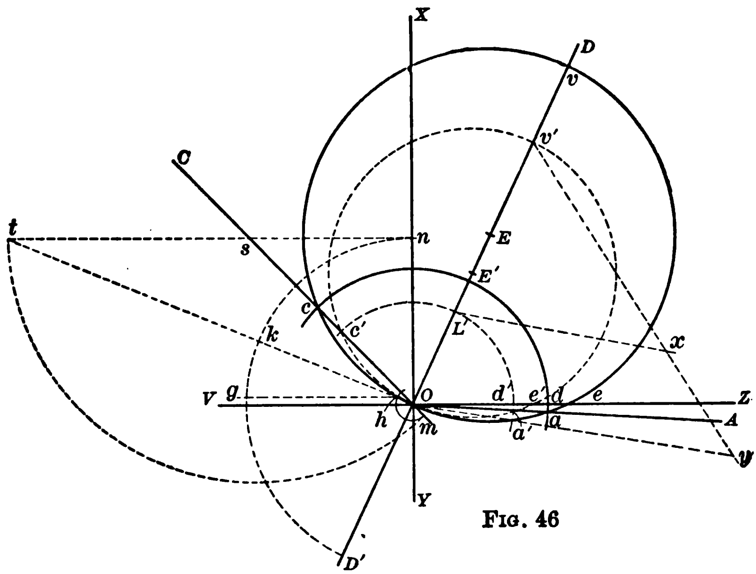



FIG. 46

opening. Through *s* draw *nst* parallel to the line of stroke *VZ*, making *st* equal to *sm* or $1\frac{1}{8}$ inches, as shown by the arc *mt*. Then join *O* and *t*, and on *Ot* take *Oh* equal to *Om* or $\frac{1}{8}$ inch, as shown by the arc *mh*. Then about *O* as a center, and with a radius equal to *On*, describe an indefinite arc. Through *h* draw *gh* parallel to *VZ*. Then *gh* will cut the indefinite arc just drawn in *g*. The arc cuts *Ot* in *k*. Then pick up the arc *kn* in the compass or dividers, and lay it off from *g* to *D'*; that is, *kn* equals *gD'*. Then join *D'* and *O*, and produce *D'O* to any point, as *D*. Then is *DOD'* the center line of the valve circles, and *DOX* is the angle of

advance. Now take any convenient radius, as $E'O$, and describe the trial valve-circle, intersecting $OD$ at $v'$. Then $Oc'$ is the corresponding lap, and by drawing in this *trial* lap-circle it will be found that the greatest possible port-opening is $L'v'$ with this lap and valve-travel. If this is not equal to $1\frac{7}{8}$ inches, the given amount—and it probably will not be,—draw $v'y$ in any convenient direction, and take $v'x$ equal to $1\frac{7}{8}$ inches, the required maximum port-opening. Then join $L'$ and $x$, and from $O$ draw $Oy$ parallel to $L'x$, until it cuts $v'y$ in $y$. Then is $v'y$ the required eccentricity. Lay off $Ov$ equal to $v'y$, and draw in the valve-circle with this diameter, which will be found to be $2\frac{3}{8}$ inches. This valve-circle will cut $OC$, the crank-position at cut-off, at $c$, and $Oc$, which is $1\frac{1}{16}$ inches, is the outside lap on the head-end. The lead is $de$, and it will be found to measure $\frac{1}{8}$ of an inch.

6. *Inside Lap, Head-End.*—This could be found on the same sheet, but by this time that drawing has become too complicated and full of lines to permit of any close determinations. Take another sheet of paper, and proceed to draw the full diagram for the head-end, as shown in Fig. 47. This drawing may be made on a smaller scale than the preceding one, if desired, but it is not desirable to make it any less than actual size. Lay off the things already determined on this sheet—that is, draw $XY$, $VZ$, $DOD'$, $OC$, the upper valve-circle whose center is $E$, and the outside lap-circle. Now turn back to Fig. 45, and locate thereon the crank-position $OT$, the crank-position corresponding to the given point of compression on the head end. (Compression occurs on the return stroke, and the crank is therefore below the line of stroke when it occurs.) Transfer $OT$ to Fig. 47, where it will intersect the lower lap circle at $t$, thus determining $Ot$, the inside lap on the head end, which will be found to measure $\frac{1}{2}$ an inch. Release and admission will take place, as shown, at the crank-positions $OA$ and $OR$.

7. *Inside and Outside Laps on Crank-End.*—These can be

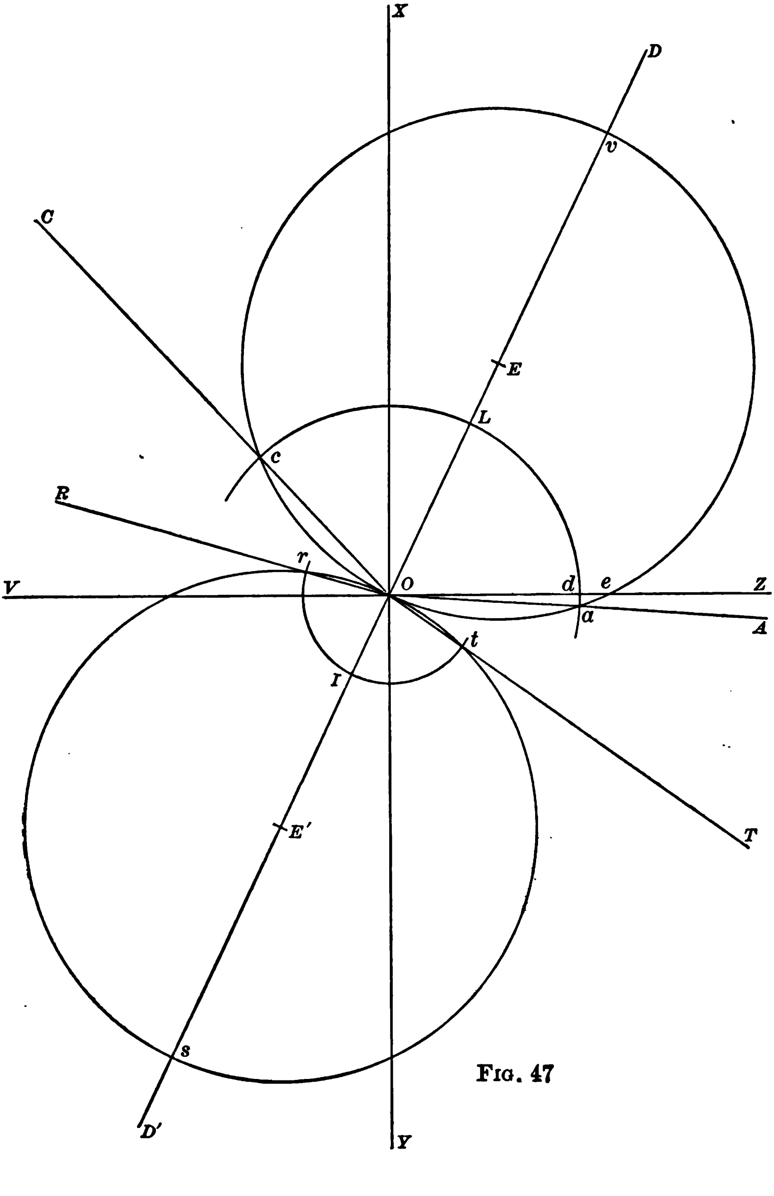



FIG. 47

found very readily on the last drawing, but for purposes of explanation they are determined separately in Fig. 48, which is an exact reproduction of Fig. 47 as far as the angle of advance and the valve-circles are concerned. Then pass back to Fig. 45 and determine $OC'$ and $OT'$, the crank-positions corresponding to cut-off and compression on the crank-end. Then transfer them to Fig. 48, and thus determine $Oc'$, the outside lap, to be $\frac{13}{16}$ inch, and $Ot'$, the inside lap, which is $\frac{5}{8}$ of an inch.

Of course in making these determinations, all the crank-positions required will be first located on Fig. 45 and marked for identification, and then transferred to Figs. 46 and 47 as needed. Fig. 48 will be dispensed with.

8. *Minimum Width of Bridge.*—In this case the valve has no overtravel, and the bridge width should be made equal to the thickness of the cylinder-walls, which are assumed in this case to be one inch.

9. *Width of Exhaust-port.*—This is determined by the method given in the preceding chapter. There are three different rules given there, and in addition to that there is a difference between the two ends of the valve, thus giving rise to the question as to which is the proper one to be used. Always employ the one which gives the greatest result, thus being sure that the port is plenty wide enough. In this case the third rule is the one to be taken, giving

Half-travel + port width − bridge + inside lap = exhaust-port width.

$$2\tfrac{7}{8} + 2\tfrac{3}{16} - 1 + \tfrac{5}{8} = 5\tfrac{5}{16}.$$

The results obtained on the preceding text may be collected as follows:

| | |
|---|---|
| Eccentricity | $2\frac{7}{8}$ |
| Valve travel (= 2 × eccentricity) | $5\frac{3}{4}$ |
| Head end, outside lap | $1\frac{1}{16}$ |
| " inside lap | $\frac{1}{2}$ |

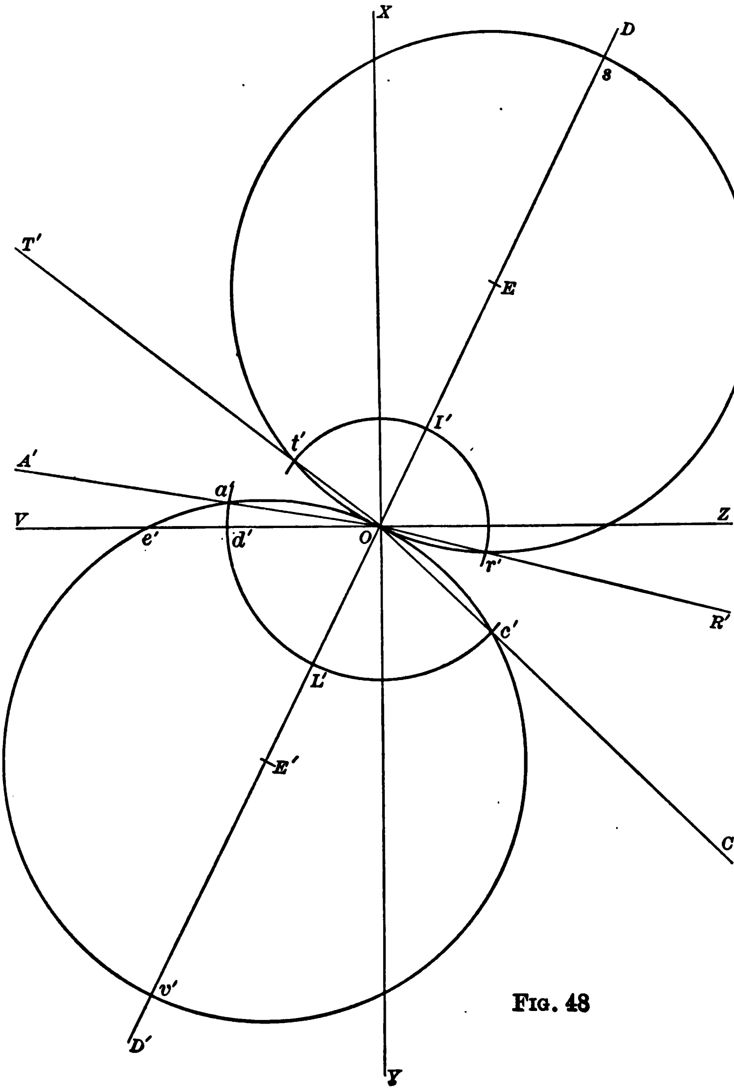



Fig. 48

| | |
|---|---|
| Crank-end, outside lap | $1\frac{3}{16}$ |
| " inside lap | $\frac{5}{8}$ |
| Exhaust-port width | $5\frac{5}{16}$ |

The next thing in order is

TO LAY OUT THE VALVE.

This is a comparatively simple matter, but one in which many beginners go astray, owing to the fact that the valve is not the same on both ends, and consequently cannot be laid out very well from a center line.

Draw the line *VZ*, Fig. 49, to represent the valve-seat.

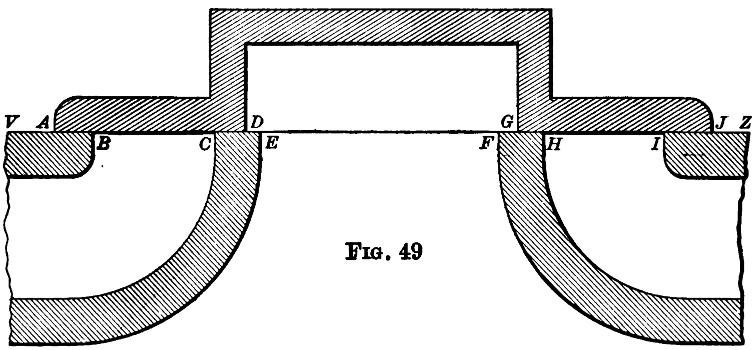



Fig. 49

Then start to lay out the valve from either end—the crank end being taken in this case. Start at any point, as *A*, and lay out the valve as follows:

| | | | |
|---|---|---|---|
| Make $AB$ = | outside lap, crank-end | $= 1\frac{3}{16}$ | inches. |
| $BC$ = | width of port | $= 2\frac{3}{16}$ | " |
| $CE$ = | " " bridge | $= 1$ | " |
| $EF$ = | " " exhaust | $= 5\frac{5}{16}$ | " |
| $FH$ = | " " bridge | $= 1$ | " |
| $HI$ = | " " port | $= 2\frac{3}{16}$ | " |
| $IJ$ = | outside lap, head-end | $= 1\frac{1}{16}$ | " |

Then draw in the section of the ports as shown, making the thickness of the seat one inch, the same as the cylinder-walls and bridge. Next,

$$\begin{aligned} \text{Make } CD &= \text{inside lap, crank end} = \tfrac{5}{8} \text{ inch.} \\ HG &= \text{inside lap, head end} \ = \tfrac{1}{2} \text{ ``} \end{aligned}$$

Then draw in the valve proper, making it thick enough to withstand the pressure to which it is subjected. The height of the cavity in the valve need not be very great, the general proportions shown in the cut being very good.

Laying out the valve-seat first and the valve afterwards avoids confusion.

# CHAPTER VIII.

## LENGTH OF VALVE-CHEST, VALVE-STEM, AND ECCENTRIC-ROD.

In addition to the dimensions of the valve and seat as determined in the preceding chapter, there are three more to be found. These are:

1. The length of the valve-chest.
2. The length of the valve-stem.
3. The length of the eccentric-rod.

These will be considered in the order given.

### LENGTH OF VALVE-CHEST.

This must be at least long enough to allow the full travel of the valve. That is, it must permit the valve to go to the two extremes of its travel as shown in Figs. 50 and 51.

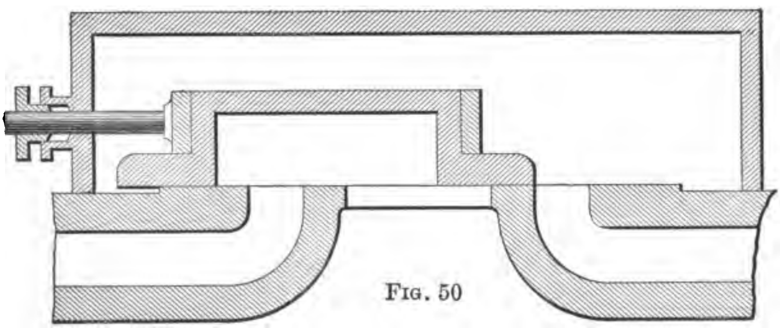


Fig. 50

The minimum length of the chest may then be measured directly from the drawing, which represents the valve in its

middle position, as in Fig. 49, by laying off the half-travel at each end of the valve, and measuring between the marks thus obtained. Or if it is desired to obtain the length of

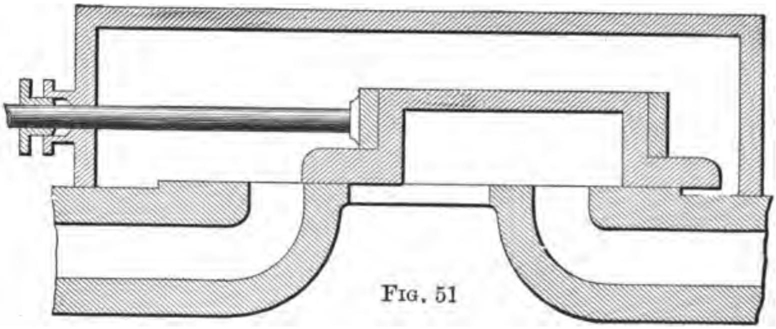


FIG. 51

valve-chest necessary without drawing the figure, it can be done by applying the following rule:

*Add together the outside laps at the two ends of the valve, twice the width of the steam-port, twice the thickness of the bridge, the width of the exhaust-port and the valve-travel. This gives the minimum inside length of the steam-chest, and it should be increased by at least one inch.*

## LENGTH OF VALVE-STEM.

One of the most common methods of connecting the valve and valve-rod or stem is shown in Fig. 52. This consists of a collar which fits closely around the valve, and may or may not be otherwise secured to it. The collar is tapped at *B*, and the valve-stem screwed in. The length of the valve-rod, *VR*, is measured from the inside of the collar to the point at which the stem is secured to the eccentric-rod, or, if a rocker is employed, to that. The length of the valve-stem must be sufficient to permit its reaching from the valve when in its extreme position, as shown in Fig. 51, to the point where

it joins either the eccentric-rod or the rocker-arm. This gives a direct means of obtaining the proper length from the drawing. The valve shown in the drawing has overtravel.

LENGTH OF THE ECCENTRIC-ROD.

Figs. 42, 43, and 44 in the sixth chapter show the determination of this length when a rocker or bell-crank is

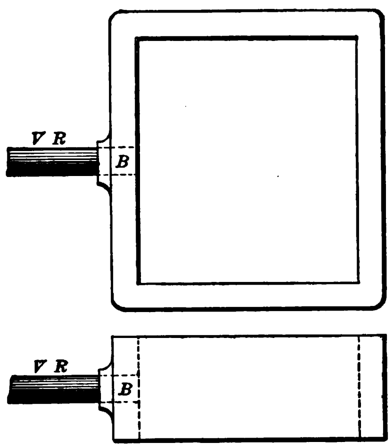



FIG. 52

employed. When neither is used, the length of the eccentric-rod may be found by this rule:

*Find the distance from the center of the exhaust-port to the center of the shaft. This may be done from the drawing. From that distance subtract the distance from the center of the exhaust-port to the end of the valve-stem, when the valve is in its middle position. The remainder is the required length of the eccentric-rod.*

This is not strictly true, but it is near enough, the difference between the length thus obtained and the correct length being negligible, as will be apparent from Fig. 53. *O* is the

center of the shaft, $P$ is the center of the exhaust-port, and $PV$ is the length of the valve-stem. The length given by the

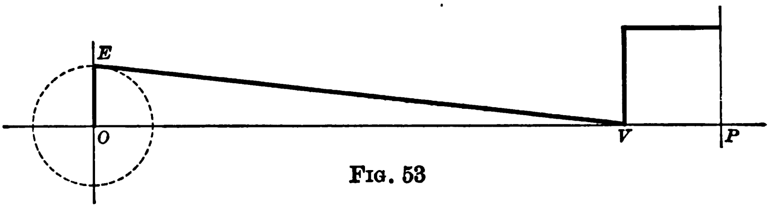



Fig. 53

rule is $OV$, whereas the correct length would be $EP$. But, for example, suppose $OE$ were 3 inches and $OV$ were 5 feet or 60 inches. Then

$$\begin{aligned} EV &= \sqrt{60^2 + 3^2} \\ &= \sqrt{3600 + 9} \\ &= \sqrt{3609} \\ &= 60.07 \text{ inches.} \end{aligned}$$

The correct length is 60.07 inches, then, against 60 obtained by the rule.

The rules for the lengths of valve-stem and eccentric-rod are given in such a way as to permit either one to be assumed and the other one found. The location of the point of suspension of a rocker, if one is employed, usually determines both of these lengths.

# CHAPTER IX.

## DESIGN OF AN ALLEN OR "TRICK" VALVE.

THE difficulty with the ordinary slide-valve is that it cannot be used for early cut-offs, on account of the large outside laps and travel required to produce the desired result. This renders the friction of the valve a very considerable item, and it is unusual to find a plain slide-valve used to secure cut-offs much earlier than five-eighths of the stroke, three-quarters being usually the limit for which they are employed.

There are, however, several valves constructed on the same general principle, which permit an early cut-off to be attained with moderate lap and travel. Prominent among these is the Allen, or "Trick," valve, shown in Figs. 54, 55, and 56, which is so constructed as to give double the amount of port-opening as a plain slide with the same amount of travel, or the same port-opening with one half the travel. This advantage is secured by means of the passage $X$ in the valve. This passage is used for admission only.

While this valve is designed to take the place of a slide-valve, it cannot be used on the same seat, but requires the face-plate shown between the slide-valve seat and the face of the Allen valve. Its operation will be best understood by reference to the figures.

The figures show the valve in its progress from one extreme of its travel to the other, and the arrows show the course of the steam. Fig. 54 represents the extreme left-hand position, Fig. 55 shows the middle position, and in Fig. 56 is shown the extreme right-hand position. With the

valve out at the left the piston is over toward the right, and steam must be admitted to the right-hand end of the cylinder through the port *P*. This is open by the amount *BE*, Fig. 54, and the steam enters through that space just as it

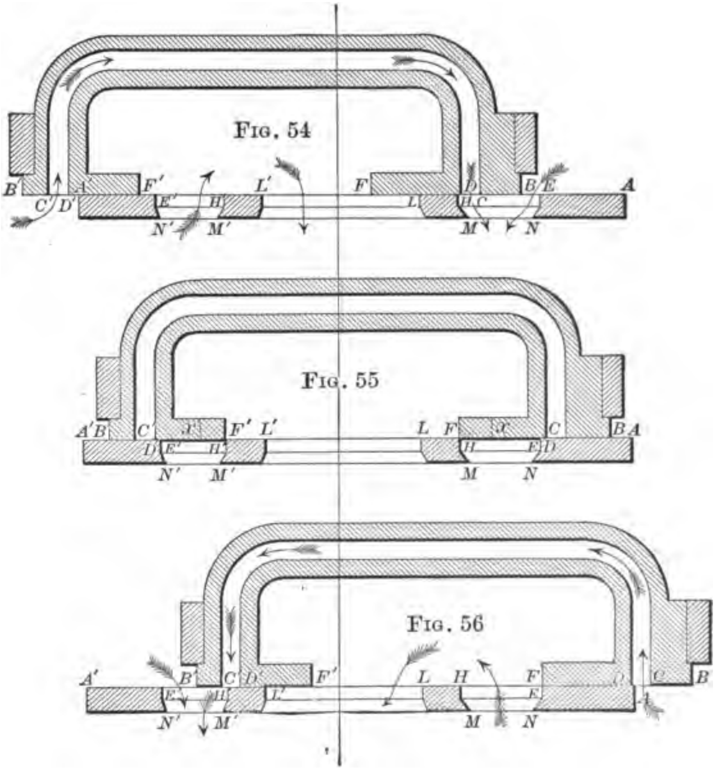


FIG. 54

FIG. 55

FIG. 56

would with an ordinary D-valve. But, in addition, the other end of the valve has moved beyond the edge of the face-plate, putting the passage *X* in communication with the steam-chest, through the opening *C'D'*, and steam is therefore admitted to the right-hand end of the cylinder through *CD*. The total opening of the port is therefore the combined amount of *CD* and *BE*. If *CD* and *BE* are equal, the total port-opening is

$2BE$; and as $BE$ is the amount that the port would be open with a plain slide-valve, it is evident that the Allen valve secures double the port-opening with the same travel. The action at the other end of the valve is the same.

Another point which must be borne in mind in regard to this valve is that it cannot be used to advantage when the cut-off is much later than one-half stroke.

In order to bring out clearly one or two points, an actual valve will be designed.

PROBLEM. To lay out an Allen valve and seat for a $12 \times 24$ engine, running at 150 revolutions per minute, connecting-rod four times the crank, cut-off on both ends at 10 inches, compression at 22 inches, lead on head-end $\frac{1}{8}$ inch. Ports to be 11 inches long.

ORDER IN WHICH THE DIMENSIONS ARE DETERMINED.

1. *Piston Speed.*—This is, by the rule given before,

$$2 \times 150 \times \frac{24}{12} = 600 \text{ feet per minute.}$$

2. *Area of Port.*—From Table IV this is found to be .15 of the area of the piston, if the velocity of steam at exhaust is 4000 feet per minute. The area of the 12-inch piston is 113.10 inches, so that the port area must be

$$113.10 \times .15 = 16.95 \text{ inches.}$$

3. *Width of Port.*—This is the area divided by the length.

$$16.95 \div 11 = 1.54 \text{ inches,}$$

or from Table II, $1\frac{9}{16}$ inches.

4. *Maximum Port-Opening.*—The velocity of steam at admission being assumed at 6000, the port-opening required is

$$\frac{4000}{6000} \times 1\tfrac{9}{16} = 1\tfrac{1}{24} \text{ inches},$$

and therefore, according to Table II, it is made $1\frac{1}{16}$ inches.

5. *Outside Lap, Angle of Advance, and Valve-Travel.*—This means the outside lap, angle of advance, and valve-travel necessary to give the required point of cut-off on the head-end for an ordinary D-valve, *but with one-half the given port-opening.* The Allen valve will double this half, giving the required amount.

First determine the crank-position corresponding to cut-off on the head-end. This operation has been fully explained in previous chapters, and will not be touched on again.

Having located this crank-position, the example then becomes a case of Problem IV, Chapter V. The diagram is given in Fig. 57, and the dimensions determined therefrom are:

| | | |
|---|---|---|
| Valve travel.................... | $4\frac{1}{2}$ | inches. |
| Eccentricity (= half-travel)...... | $2\frac{1}{4}$ | " |
| Outside lap, head-end ........... | $1\frac{3}{4}$ | " |

6. *Inside Lap, Head-End.*—This is determined from the diagram, Fig. 58, where $T$ is the crank-position with the piston at 22 inches. The inside lap is, as explained in Chapter III, that part of the crank-position included between the valve-circle and the point $O$ of the diagram. The inside lap and outside lap are read from different valve-circles. It will be noticed that the line $OT$ does not cut the lower valve-circle below the line of stroke, but above it. The inside lap is therefore negative; that is, the valve has inside clearance. While this is a possible construction, it is not at all desirable, as there would evidently be a time in the stroke when the two ends of the cylinder would be in direct communication with

each other and with the exhaust. This will be plain on reference to Fig. 55, where the dotted lines $x\,x'$ show the construction of a valve with inside clearance at both ends. If only one end has this negative inside lap, the blow-through of steam would not be in evidence with the valve in its middle position, but will appear later, unless the other end has a con-

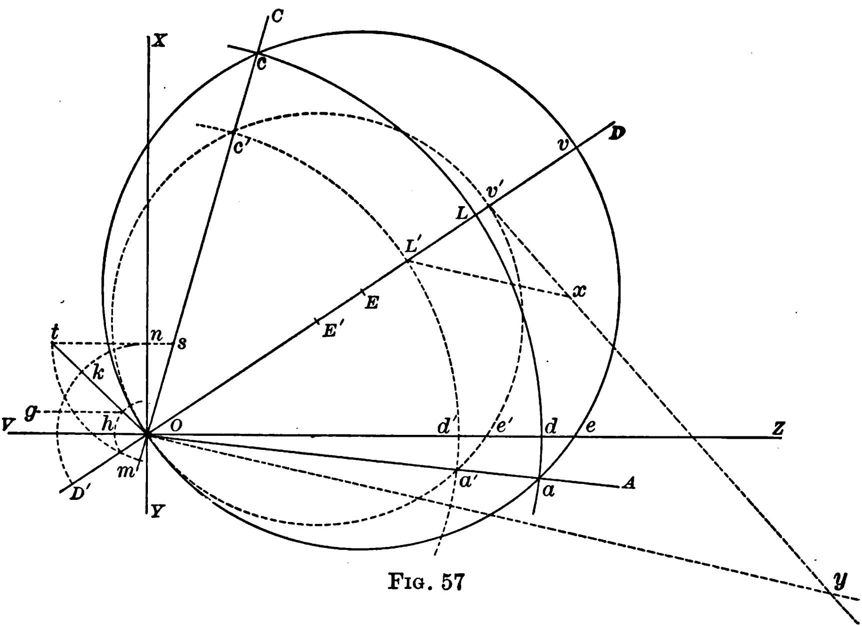



FIG. 57

siderable amount of inside lap, at least equal to the inside clearance of the first end.

The inside lap on the head-end will therefore be made zero.

7. *Inside and Outside Laps, Crank-End.*—These are found from the diagram, in the manner already explained. The outside lap is $Oc'$, Fig. 58, which is $1\frac{5}{16}$ inches. The inside lap is again negative, as shown by $OT'$ cutting the upper valve-circle below the line of stroke. The inside lap will therefore be made zero.

For the rest of the dimensions the description will refer to the valve in its middle position. as in Fig. 55.

8. *The Distances DE and D'E'.*—These are equal and may be assumed, and in this case will be made $\frac{1}{8}$ inch.

9. *Width of Port, MN and M'N'.*—These being the open-

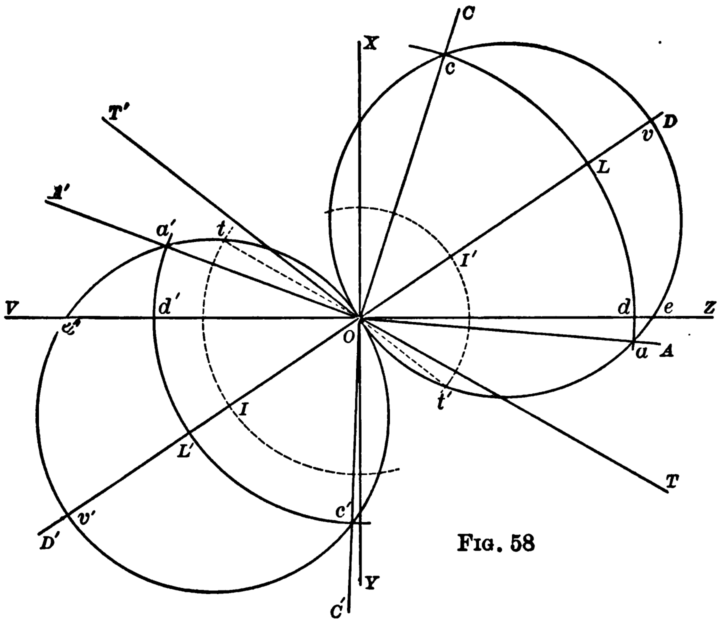



FIG. 58

ings in the cylinder-face, they are made equal to the port width determined, or $1\frac{9}{16}$ inches each.

10. *Width of Passage in Valve, or CD and C'D'.*—These are each made equal to one-half the port-opening. The port-opening being $1\frac{1}{16}$, the half is $1\frac{17}{32}$, and *CD* and *C'D'* will be made the next larger sixteenth, or $\frac{9}{16}$.

These last three are all the dimensions that are the same on both ends of the valve.

11. *BE* is made equal to the outside lap on the head-end, assuming the cylinder to be at the right. $BE = 1\frac{3}{4}$ inches.

12. *B'E'* is made equal to the outside lap on the crank-end, or $1\frac{5}{16}$ inches.

13. *AC*. This is equal to the outside lap, head-end, $1\frac{3}{4}$ inches.

14. *A'C'*. This is equal to the outside lap on the crank-end, $1\frac{5}{16}$ inches.

15. *BC*. This is made equal to the outside lap, head-end —one-half the port-opening—*DE*. This gives $1\frac{3}{4} - \frac{9}{16} - \frac{1}{8} = 1\frac{1}{16}$ inches.

16. *B'C'*. This is made equal to the outside lap, crank end—one-half the port-opening—*D'E'*. This gives $1\frac{5}{16} - \frac{9}{16} - \frac{1}{8} = \frac{5}{8}$ inches.

No allowance is made for the difference between the port-openings at the two ends of the cylinder, because the crank-end opening, as shown by the diagram, is greater than that of the head-end. Therefore, if the dimensions be made large enough to permit the required opening on the head-end, the crank-end opening will be ample with the same allowance.

The next thing is to describe the method of

## LAYING OUT THE FALSE SEAT *AA'*.

17. *Thickness*.—This may be made anything desirable.

18. *AE*. This is made equal to the outside lap, head-end + one half the port-opening + *DE*; or, in this case, $1\frac{3}{4} + \frac{9}{16} + \frac{1}{8} = 2\frac{7}{16}$ inches.

19. *A'E'*. This is calculated in a similar manner: Outside lap, crank-end + one half of the port-opening + *D'E'*, which gives $1\frac{5}{16} + \frac{9}{16} + \frac{1}{8} = 2$ inches.

20. *EH*. This made equal to the total port-opening + *BC*, or $1\frac{1}{16} + 1\frac{1}{16} = 2\frac{1}{8}$ inches.

21. *E'H'*. In a similar way this is made equal to the total port-opening + *B'C'*, which gives $1\frac{1}{16} + \frac{5}{8} = 1\frac{11}{16}$.

22. *Width of Bridge*.—This is made equal to the half-travel—(*EH* or *E'H'*)—inside lap + $\frac{1}{4}$ inch, for a minimum.

$EH$ or $E'H'$ is used, according to which gives the greater value. The inside lap to be used is determined in the same way. $E'H'$ is the value for this case, and the inside lap is zero at both ends, so that the rule gives $2\frac{1}{4} - 1\frac{11}{16} + \frac{1}{4} = \frac{13}{16}$ inches. This is the minimum thickness of the bridge, and if it is any greater than the thickness of the rest of the casting, the bridge must be thickened to the required amount. If not, the bridge may be made of the same thickness as the rest of the cylinder-casting. This will be done in this case, and the bridges $HL$ and $H'L'$ will each be made one inch.

23. *Width of Exhaust-port.*—This is made at least equal to the half-travel + width of steam port − bridge + inside lap; and the width will then be, in this case, at least $2\frac{1}{4} + 1\frac{1}{16} - 1 + 0 = 2\frac{5}{16}$; and it may be anything greater.

All the dimensions of the valve, valve-seat, and false seat are now determined, and the valve may be laid out in its middle position, as shown in Fig. 55, beginning at either end.

## CHAPTER X.

### DESIGN OF A DOUBLE-PORTED VALVE.

THE valve shown in Figs. 59–63, known as the "double-ported" valve, is another type of valve used to replace a D valve when it is desired to secure an early cut-off; and it does this in the same general manner as the Allen valve, by securing the same port-opening as a D-valve with one-half the travel, or double the opening with the same travel. This enables the same point of cut-off to be secured with a smaller amount of outside lap. The lap and travel being reduced, it follows that the friction of the valve is lessened. From Figs. 60 and 61 it will be seen that the steam enters the cylinder beneath the outer edges of the valve, and that the action of the outer shell is therefore similar to that of a plain D-valve. In addition to this means of admission, passages $S$ and $S'$ are provided which run all the way through the valve, and are therefore open to live steam from the steam-chest at all times. These passages have ports in the bottom, as shown at $GH$ and $G'H'$. There are corresponding ports $IL$ and $I'L'$ in the valve-seat, and steam is thus admitted through these secondary ports as shown by the arrows in Figs. 60 and 61. If the valve as shown in Fig. 60 is in its extreme left-hand position, the port opening to steam is the sum of $G'H'$ and $A'C'$. The opening $A'C'$ is that which would be obtained by a plain D-valve, and therefore, if the valve is so constructed that $A'C'$ and $G'H'$ are equal, the total port-opening is double that which would be obtained by a plain slide-valve.

The exhaust passes through the passage $X$, as shown by

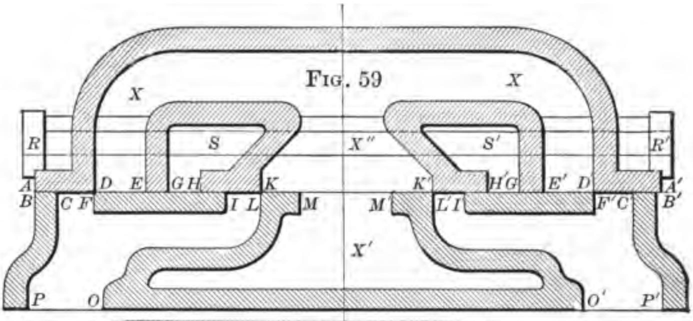


Fig. 59

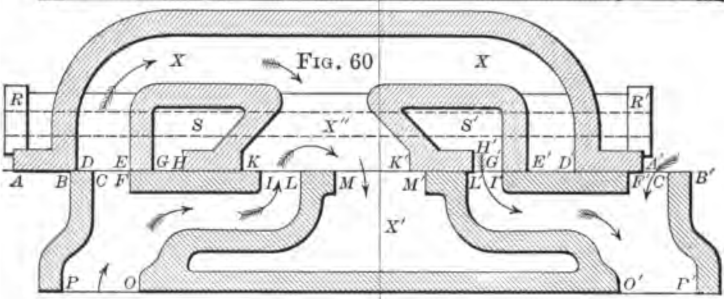


Fig. 60

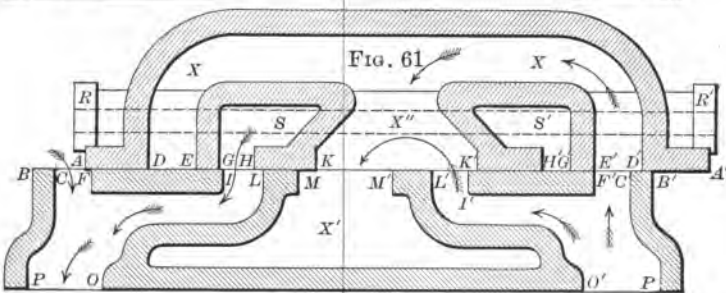


Fig. 61

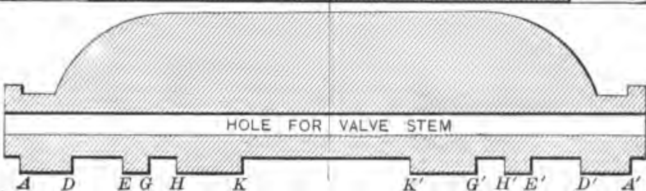



**Fig. 62**

the arrows; and this passage gives the valve a certain similarity to the Allen valve; but here the similarity ends, for it will be remembered that with the Allen valve the passage is used for the admission alone, while here it is only for the exhaust.

The action of this valve is therefore the same as would be obtained by two slide-valves, one within the other. The difference in construction is that the exhaust-passage, $X''$, of the inner valve is thrown open to the exhaust-passage, $X$, of the outer valve. It will be noticed that $S$ and $S'$ are used for admission only. They are not of the same size throughout, but are greater at the sides, where they open to the steam-chest, as shown clearly in Fig. 63, in which the left half

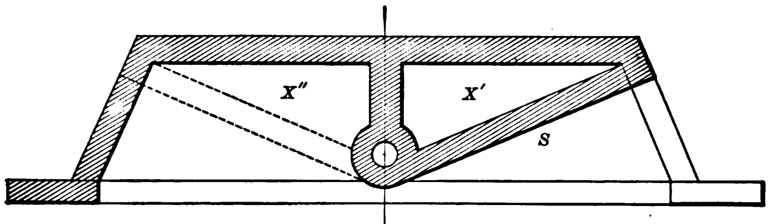



FIG. 63

of the figure shows a transverse section through the middle of the valve, and the right-hand half is a section through the center of the passage $S$. This figure also shows the general shape of the passage $X$, which is divided into two parts by the central web. Fig. 62 shows a section through the center of the valve, Figs. 59, 60, and 61 being some distance to one side.

This type of valve is more often applied to vertical engines than to any others, and is designed to give equal cut-offs, for the following reasons. With equal cut-offs secured by equal laps the leads are unequal, and the greatest lead is at the crank-end of the cylinder, as shown by the diagrams in Chapter III. The crank-end of the cylinder being the lower

end, the extra amount of lead is useful in cushioning the piston and piston-rod, etc., which must, from the nature of the construction, have a greater force on the down than on the up stroke. Another reason, or, rather, one in the same line, for not designing the valve to have equal leads, is that that would give a longer cut-off on the crank-end, which is the end where the weight of the moving parts tends to increase their motion rather than retard it.

This valve, unlike the Allen, cannot be used on the same valve-seat as a plain D valve by the interposition of a false seat, but requires the valve-seat to be laid out with reference to the double-ported valve. It is therefore impossible to take off a D-valve from an engine and put a double-ported valve in place.

To show the process of designing a valve, an actual case will be worked out.

## PROBLEM.

GIVEN. A cylinder $27 \times 24$ inches; ports 24 inches long; revolutions per minute, 140; width of bridge, $1\frac{3}{4}$ inches; inside lap, both ends, 0; cut-off, both ends, $\frac{3}{4}$ stroke; connecting-rod, 72 inches; velocity of admission, 6000 feet per minute; velocity of exhaust, 4000 feet per minute; lead, head end, $\frac{1}{16}$ inch.

REQUIRED. Outside lap, both ends; travel of valve; lead, crank-end; maximum port-opening, both ends; and to lay out the valve in its middle position as shown in Fig. 59.

### ORDER IN WHICH THE DIMENSIONS ARE DETERMINED.

1. *Piston Speed.*—This is

$$140 \times 2 \times \frac{24}{12} = 560 \text{ feet per minute.}$$

2. *Area of Steam-Port.*—The nearest piston speed, to the given 560, in Table IV is 550, and the port area is given as .138 times the piston area for a velocity of 4000 feet per minute. The piston area from Table V is 572.56 square inches, so that the port area is

$$572.56 \times .138 = 79.01.$$

3. *Width of Steam-Port.*—The ports being 24 inches long, the width must be

$$79.01 \div 24 = 3.3.$$

This is, however, the *total* width of port at one end of the cylinder, and will be taken as $3\frac{1}{2}$ inches. There are two ports at each end of the cylinder, so that the width of each one is made $3\frac{1}{2} \div 2 = 1\frac{3}{4}$ inches. This determines $I'L'$, $C'F'$ at the head-end, and $IL$ and $CF$ at the crank-end, each being made $1\frac{3}{4}$ inches. The width of the main port—that is, after the two ports join—must be made $3\frac{1}{2}$ inches.

4. *Maximum Port-Opening on the Head-End.*—This is the opening which will give the entering steam the required velocity, and is found as follows:

$$\text{Port-opening} = \text{port width} \times \frac{\text{velocity of exhaust}}{\text{velocity of admission}}.$$

This gives:

$$\text{Port-opening} = 1\frac{3}{4} \times \frac{4000}{6000} = 1\frac{3}{16}.$$

This is the opening of each port, and the total opening at one end of the cylinder is therefore $2 \times 1\frac{3}{16} = 2\frac{3}{8}$.

The same result would be obtained by using Table IV to find the area required for free admission.

5. *Valve-Travel, Inside and Outside Laps on Both Ends, and Maximum Port-opening and Lead on Crank-End.*—These are determined precisely as for the Allen valve, and the explana-

tion given of Fig. 57 will answer for this, the numerical values only being changed. The results obtained are as follows:

| | |
|---|---|
| Outside lap, head end................ | $1\frac{1}{4}$ inches. |
| Outside lap, crank end............... | 1 " |
| Lead, crank end..................... | $\frac{3}{8}$ " |
| Maximum port-opening, crank end.... | $1\frac{1}{2}$ " |
| Valve travel......................... | 5 " |

6. *Width of Bridge.*—The minimum width allowable is

Half-travel − width of one port − inside lap + $\frac{1}{4}$ inch.

$$2\frac{1}{2} \quad - \quad 1\frac{3}{4} \quad - \quad 0 \quad + \frac{1}{4} = 1 \text{ inch.}$$

This is less than the $1\frac{3}{4}$ inches given in the problem. This rule is only applied to see whether the width given is sufficient to prevent blowing through. The result obtained shows that the $1\frac{3}{4}$ inches, which is the thickness of the cylinder-casting, is ample.

7. *Width of GH.*—This is made equal to the port-opening for that end of the cylinder, and is therefore $1\frac{1}{2}$ inches. *GH* is at the crank end.

8. *Width of G'H'.*—Make this equal to the port-opening at its end of the cylinder, or $1\frac{3}{16}$ inches, *G'H'* being at the head end.

9. *Width EG.*—This may be made any convenient figure, and in this case it will be one inch.

10. *Width E'G'.*—This is made equal to *EG*.

11. *Width of Exhaust-Port.*—This must be made great enough so that when the valve is in either of its extreme positions, as shown in Figs. 60 and 61, the opening to exhaust is so great that the outflowing steam will not have a greater velocity than 4000 feet per minute. The *total width of steam-ports* at one end is that found to be necessary for this velocity, and, as the exhaust-port takes care of the exhaust from both

of these ports, the free opening of the exhaust $MK'$, Fig. 60, or $KM'$, Fig. 61, must be equal to the total width of the ports at one end of the cylinder. The exhaust-port, $MM'$, is then made of a width equal to

Half-travel + inside lap − bridge + port width at one end.

$2\frac{1}{2} + 0 - 1\frac{3}{4} + 3\frac{1}{2} = 4\frac{1}{4}$ inches.

12. *Width FI.*—The edge $E$ must not travel beyond $F$ if it is desired to have a full opening during exhaust. That gives the value of $FI$ as

Outside lap, crank-end + $GH$ + $EG$ + half-travel.

$1 + 1\frac{1}{2} + 1 + 2\frac{1}{2} = 6$ inches.

13. *Width F′I′.*—This is obtained in a similar manner to $FI$; or it is equal to

Outside lap, head-end + $G'H'$ + $E'G'$ + half-travel.

$1\frac{1}{4} + 1\frac{3}{16} + 1 + 2\frac{1}{2} = 5\frac{15}{16}$.

The edge $E$ may be allowed to overtravel very slightly the point $F$, in case it is an object to shorten the valve. Doing so prevents the exhaust from being wide open all the time, but the choking only occurs during a small part of the stroke. When this overtravel is allowed, $FI$ and $F'I'$ are to be shortened by the amount allowed. No allowance is made in this case.

14. *Width DE.*—This is made equal to

Half-travel, − inside lap, crank-end − amount that $E$ overtravels $F$.

$2\frac{1}{2} - 0 - 0 = 2\frac{1}{2}$ inches.

15. *Width D′E′.*—This is made equal to

Half-travel − inside lap, head end − amount $E'$ overtravels $F'$.

$2\frac{1}{2} - 0 - 0 = 2\frac{1}{2}$ inches.

The inside laps being equal, it happens that $DE$ and $D'E'$ are equal. If the inside laps are unequal, the values obtained by the rules would be unequal; but both $DE$ and $D'E'$ would be made equal to the larger value obtained, keeping the valve symmetrical in this respect.

16. *Width BC.*—This must be so long that $D$ will not overtravel $B$; for if it did, the steam-chest would be in direct communication with the exhaust-passage $X$, allowing steam to blow straight through the valve. Its width must therefore be at least equal to

| Half-travel | — width of one port | — inside lap, crank-end | + ¼ inch. |
|---|---|---|---|
| $2\frac{1}{2}$ | $-\ 1\frac{3}{4}$ | $-\ 0$ | $+\frac{1}{4} = 1$ inch. |

17. *Width B′C′.*—This is determined, from the same considerations that govern the dimension $BC$, to be

| Half-travel | — width of one port | — inside lap, head-end | + ¼ inch. |
|---|---|---|---|
| $2\frac{1}{2}$ | $-\ 1\frac{3}{4}$ | $-\ 0$ | $+\frac{1}{4} = 1$ inch. |

The next thing in order is

## TO LAY OUT THE VALVE.

18. *Tabulate the Dimensions.*—The dimensions of the valve and seat should be tabulated separately, in the order in which they are to be laid down, beginning at either end. In this case the start will be made from the crank end. The letters apply to Fig. 59.

*Valve Seat.*

| | | | | | | |
|---|---|---|---|---|---|---|
| *BC.* | 1 | inch | ............obtained from | | | 15 |
| *CF.* | $1\frac{3}{4}$ | " | ............ | " | " | 3 |
| *FI.* | 6 | " | ............ | " | " | 11 |
| *IL.* | $1\frac{3}{4}$ | " | ............ | " | " | 3 |
| *LM.* | $1\frac{3}{4}$ | " | ............ | " | " | 6 |

| | | | | |
|---|---|---|---|---|
| *MM'*. | $4\frac{1}{4}$ | inch | ............obtained from | 10 |
| *M'L'*. | $1\frac{3}{4}$ | " | ............ " " | 6 |
| *L'I'*. | $1\frac{3}{4}$ | " | ............ " " | 3 |
| *I'F'*. | $5\frac{15}{16}$ | " | ............ " " | 12 |
| *F'C'*. | $1\frac{3}{4}$ | " | ............ " " | 3 |
| *B'C'*. | 1 | " | ............ " " | 16 |

Lay this out on any convenient scale.

*The Valve.*

| | | | | | |
|---|---|---|---|---|---|
| *AC*. | 1 | inch | outside lap | ........obtained from | 5 |
| *DF*. | 0 | " | inside lap | ........given " | |
| *DE*. | $2\frac{1}{2}$ | " | | ... ....obtained " | 13 |
| *EG*. | 1 | " | | ........ " " | 8 |
| *GH*. | $1\frac{1}{2}$ | " | | ........ " " | 6 |
| *HK*. | 1 | " | outside lap | ...... " " | 5 |
| *LK*. | 0 | " | inside lap | .... ...given | |
| *K'L'*. | 0 | " | " " | ........ " | |
| *I'H'*. | $1\frac{1}{8}$ | " | outside lap | ........obtained from | 5 |
| *H'G'*. | $1\frac{8}{15}$ | " | port opening | ...... " " | 7 |
| *G'E'*. | 1 | " | | ........obtained from | 9 |
| *E'D'*. | $2\frac{1}{2}$ | " | | ........ " " | 14 |
| *F'D'*. | 0 | " | inside lap | ........given | |
| *C'A'*. | $1\frac{1}{4}$ | " | outside lap | ........obtained from | 5 |

Reference to Fig. 63 will show that the exhaust-passage $XX$, Fig. 59, is divided into two portions, $X'$ and $X''$, when it passes over either of the steam-passages, as $S$. Care must be taken to make the valve of so great a height that the sum of the areas of these two portions shall be at least as great as the total area of the ports at one end of the cylinder in order to maintain the proper velocity of exhaust.

# CHAPTER XI.

## VALVE-SETTING.

HAVING designed a valve and having it in place on the engine, the next thing to be considered is the proper method of setting it in order to secure the best results. The method to be employed will depend on the design of the valve—whether it is intended to secure equal cut-offs on the two ends, or whether it was made to give equal leads; or, again, whether the cut-offs and leads were equalized, according to the method given in Chapter VI.

In any case the first step is to put the engine on the center, by which is meant that the piston is at the end of its stroke and the crank and connecting-rod are in the same straight line. This operation must be performed very precisely, as little variation in the position of the crank from the line of stroke or piston from the end of the stroke will make a large difference in the position of the valve. This is because the eccentric is at or near its middle position at the time the crank is on the center. The crank-pin is moving vertically while the eccentric is moving horizontally at that instant, so that a small angular movement of the eccentric will make a great difference in the position of the valve. The crank, however, is moving vertically, and a comparatively large angular movement of that will only move the cross-head a little ways from the end of its travel. Consequently the dead-center must be located very exactly or the valve-setting will be thrown out.

TO PUT AN ENGINE ON THE CENTER.

The engine is put on the center by moving the cross-head a measured distance on each side of its extreme travel and

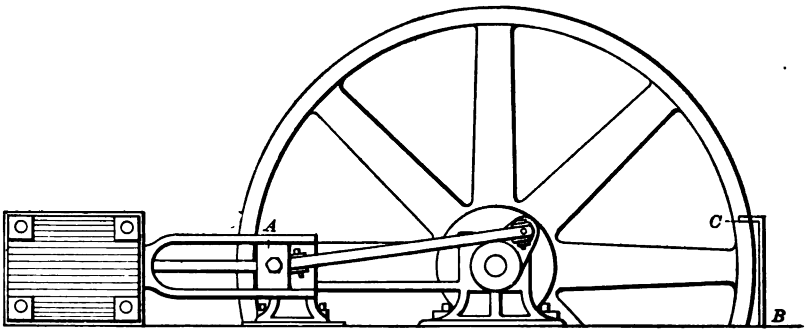



FIG. 64

measuring the amount of the movement of the fly-wheel by means of marks on the rim. This distance is bisected, and

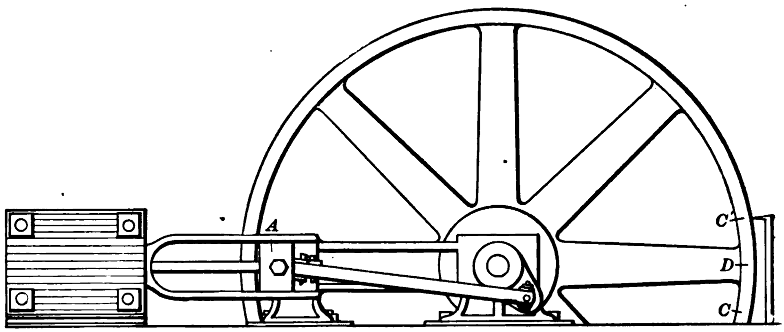



FIG. 65

the middle point determines the true center. The practical method of doing this is shown in Figs. 64, 65, and 66.

*First.* Turn the engine in the direction in which it is to run until the cross-head is nearly at the end of its stroke, as shown in Fig. 64.

*Second.* With the cross-head in this position, take a piece of chalk and make a mark across the cross-head and guides, as

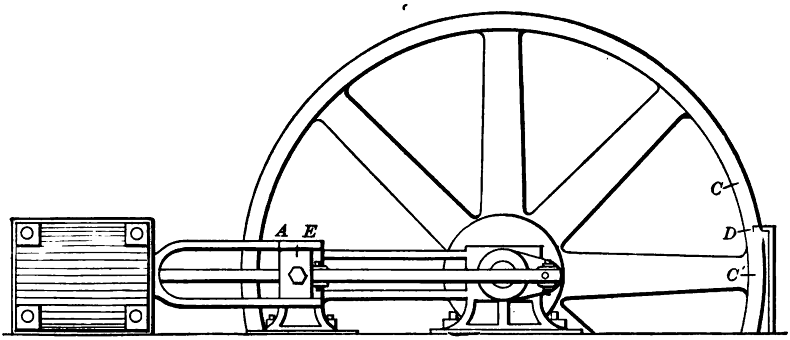



Fig. 66

shown at *A*, Fig. 64. This mark may be put anywhere on the cross-head, as it serves only as a reference-mark.

*Third.* Fix an upright pointer as near as possible to the face of the fly-wheel, as shown at *B*, and mark the height reached by its end, as shown at *C*.

*Fourth.* Turn the engine, still in the direction in which it is to run, until the mark on the cross-head again comes even with the mark on the guides, as shown in Fig. 65.

*Fifth.* Make another chalk-mark on the fly-wheel opposite the end of the upright, as shown at *C'*, Fig. 65. The position of the first mark after this movement is then as shown at *C*. The distance *CC'* then represents the amount that the fly-wheel has moved, while the cross-head has moved from *A* out to the dead-center and back again to *A*. That is, the cross-head movement has been twice the distance from *A* to the dead-center. Consequently, if the wheel revolved until the pointer comes midway between *C* and *C'*, the engine will be on the dead-center.

*Sixth.* Find the point midway between *C* and *C'*. This is *D*, Fig. 65.

*Seventh.* Turn the engine, still in the direction in which it is to run, until the mark *D* comes opposite the end of the pointer, as shown in Fig. 66. The engine is then on the dead-center.

*Eighth.* With the engine on the dead-center, make another chalk-mark on the guides opposite the one on the cross-head, as at *E*, Fig. 66. This is to serve as a reference-mark, and the first mark is then to be erased.

The dead-center at the other end is found in the same manner.

When putting the engine on the center, or when performing any of the other operations of valve-setting, the engine must always be turned in the direction in which it is to run. This is because any lost motion, back-lash, or play in the moving parts will then affect the valve in setting precisely as under running conditions. The wheel must never be turned beyond the required point and then back to it, as the lost motion would allow a considerable movement of the fly-wheel to take place without a corresponding motion of the cross-head; and the valve-setting would then be thrown out of true.

## TO SET A VALVE FOR EQUAL LEADS.

### *First Method.*

*First.* Put the engine on the dead-center at one end of its stroke, using the method just described.

*Second.* Give the eccentric as nearly as possible the proper amount of angular advance, as determined from the valve diagrams, taking care that the amount given shall be more, rather than less, than the required amount.

*Third.* Adjust the length of the valve-stem or eccentric-rod until the lead at the end of the stroke for which the adjustment is then being made is equal to the required amount.

*Fourth.* Turn the engine to the other dead-center.

*Fifth.* Measure the lead at that end. If the measurement is the same as that at the first end, the eccentric is in its proper place, and it only remains to secure it there; but the chances are that the leads will be unequal, and in that case the next step is the

*Sixth.* Correct half the difference in the leads by changing the length of the valve-stem.

*Seventh.* Correct the remaining half of the difference by moving the eccentric. It will be apparent at once, from the nature of the difference, in which direction the eccentric is to be moved.

*Eighth.* Turn back to the other center and measure the lead. If it is the required amount, the setting is complete. If not, repeat operations *Sixth* and *Seventh* until the leads are equal.

*Ninth.* Secure the eccentric in place.

When a valve-gear has a rocker, the latter is usually designed to swing to an equal angle on each side of the perpendicular, and in any case the length of the valve-stem must be such that the rocker will move as designed. This being so, it is evident that in performing the sixth and seventh operations—those of correcting the variation in leads at the two ends of the cylinder—the length of the valve-stem must be changed very little, if any, making the sixth operation very small, and the greater part of the adjustment must be made in the seventh operation, that of changing the position of the eccentric.

### *Second Method.*

This method is a convenient one when it is difficult to turn the engine over. It is applicable only to that class of valves having harmonic motion. Harmonic motion is such as that of the foot of a perpendicular from the crank-pin upon the line of stroke when the motion of the crank-pin is uniform. This is the case in an ordinary engine, and an ordinary slide-

valve has harmonic motion. Among the valve-gears in which the valve does not have harmonic motion may be mentioned a slide-valve having equal lead and the cut-offs equalized by means of a rocker or bell-crank lever, and the link motion and radial gears.

When the motion is harmonic, the maximum port-openings will be equal when the leads are equal.

*First.* Loosen the eccentric on the shaft.

*Second.* Turn the eccentric until it gives the maximum port-opening, first at one end and then at the other.

*Third.* If the maximum port-openings are not equal—and the chances are that this will be the case—make them so, by changing the length of the valve-stem by half the difference, thus adjusting the length of the valve-stem.

*Fourth.* Put the engine on the center. This is the only time that it is necessary to perform this operation.

*Fifth.* Turn the eccentric to give the proper lead, thus adjusting the angle of advance.

*Sixth.* Secure the eccentric in place.*

TO SET A VALVE FOR EQUAL CUT-OFFS.

*First.* Put the engine on one dead-center—say the head-end.

*Second.* Give the eccentric, as nearly as can be judged, the angle of advance determined by the valve-diagram; taking care that if there is any difference, it shall be in excess of the proper amount rather than less.

*Third.* Give the valve the correct amount of lead, as nearly as possible.

*Fourth.* Move the engine in the direction in which it is to run until cut-off occurs.

*Fifth.* Measure the distance that the cross-head has moved,

*The author wishes to acknowledge his indedbteness to Peabody on Valve-gears for this method.

up to this point, from the end of the stroke. This is found by means of the chalk-marks on the guides and cross-head.

*Sixth.* Turn the engine in the direction in which it is to run until cut-off occurs on the return stroke.

*Seventh.* Measure the travel of the cross-head from the beginning of the return stroke up to cut-off. If this is the same as on the forward stroke, the valve is set correctly, and the eccentric should then be secured in place. But it is hardly probable that this result will be secured on the first trial; and in that case the next operation is the

*Eighth.* Correct the difference in cut-offs by changing the length of the valve-stem. If the cut-off is earlier on the crank-end or return stroke, the valve-stem should be lengthened. If it is earlier on the head-end or forward stroke, the valve-stem should be shortened.

*Ninth.* Put the engine on the head-end center again, and adjust the lead by moving the eccentric.

*Tenth.* Test the cut-offs and see whether or not they are equal. If they are, the adjustment is finished. If not, repeat the *eighth* operation until they are equal, and then—

*Eleventh.* Secure the eccentric in place.

As pointed out in Chapter III, designing a valve for equal cut-offs will make the leads unequal. At the time the valve is designed the lead can be measured at each end of the cylinder, and the operation of setting the valve can be performed by using the first method given above for equal leads, except that the leads, instead of being made equal, are made equal to the required amounts. In addition, the travel of the cross-head from the beginning of the stroke up to cut-off must be determined and the setting completed by the *eighth* and *ninth* operations just given.

By means of any of these methods, it is assured that the action of the valve shall be just as intended at admission or cut-off. This also assures the fact that any irregularity of action, or error of design, caused by neglecting the angularity

of the eccentric-rod, will not affect the valve while either opening or closing, but will make itself felt while the port is either opened or closed; which is of no particular consequence.

Now, having set the valve, it is of importance to make a distinct set of reference-marks on the eccentric, so that it will be possible to set it in place again very readily if it slips, thus avoiding vexatious delays, or, perhaps, the necessity of going over the whole ground cf valve-setting again.

When any one of the preceding methods of valve-setting is employed, it is necessary to remove the cover of the steam-chest in order to observe the action of the valve, measure leads, etc.

## TO SET A VALVE WITH THE CHEST-COVER ON.

After having once set the valve for equal leads, with the cover off, it is possible to arrange things so that thereafter the valve can be set with the chest-cover on, and, if absolutely necessary, with steam on. This method is illustrated in Figs. 67 and 68.

After having set the valve, the engine is placed on the center, and a reference-mark made on the valve-stem somewhere outside of the stuffing-box, as shown at *A*, Fig. 67. Then another mark is made somewhere on the chest-cover, as at *B*. Next a piece of heavy wire is taken, and a tram or spanner, *T*, Fig. 67, is made, of such length that it will reach from *A* to *B*. Then the engine is put on the other center, which operation will change the distance of the mark on the valve-stem from the mark on the chest-cover to *AB*, Fig. 68. Another spanner, *T'*, is then made which will reach that distance. Then, when it is again necessary to set the valve, the operation can be performed by employing the first method for equal leads, using the trams to determine the equality of the leads, as follows:

*First.* Put the engine on one center, say the head-end.

*Second.* Give the eccentric the proper amount of angular advance, as nearly as possible, making it too great rather than too little.

*Third.* Adjust the length of the valve-rod until the lead at the head-end is the proper amount; that is, until the tram *T* reaches from *A* to *B*.

*Fourth.* Turn the engine to the other center.

*Fifth.* Measure the crank-end lead. This is done by using the tram *T'*. Remember that when it reaches from *A* to *B*,

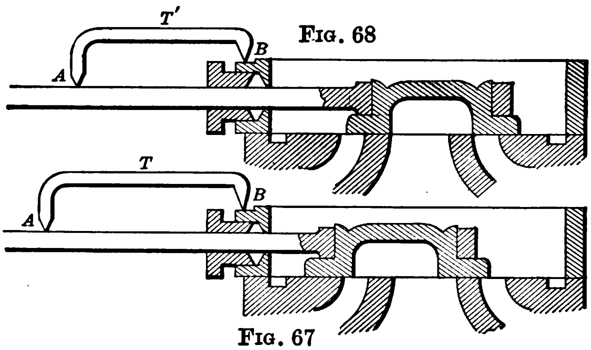



FIG. 68

FIG. 67

as in Fig. 68, the leads are equal. When it reaches to any other point, the lead is changed by the distance from *A* to the point which the tram marks on the valve-stem. If the tram spans over to the crank side of *A*, the lead is too great at the crank-end. If the tram reaches to a point on the cylinder side of *A*, the lead at the crank-end is too little.

The rest of the operations are the same as given for equal leads.

A variation of this method consists in using but one tram, and making two marks on the valve-stem, at the points reached by the tram with the engine on the two centers. The method of setting is then the same. This is frequently used on locomotives.

## CHAPTER XII.

### SHAFT-GOVERNORS. GENERAL PRINCIPLES AND TYPES.

IT was shown in the first chapter that in order to reverse the direction of rotation of an engine it is necessary to move the eccentric around the shaft past the crank until it makes the same angle with it on the opposite side that it did in its original position. That is, if the arrangement of crank and eccentric shown in Fig. 69 will cause the engine to run over,

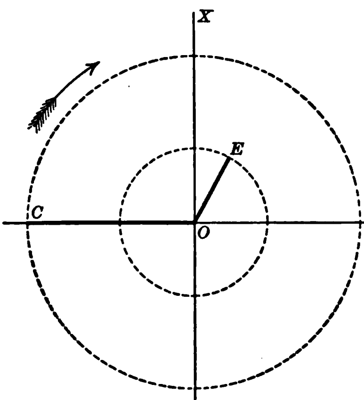


FIG. 69

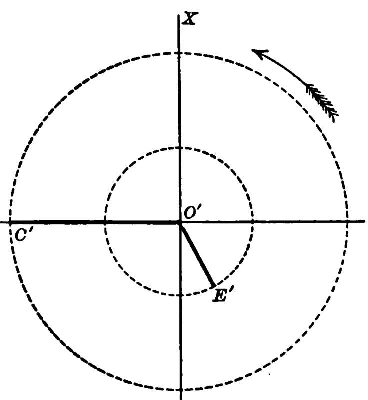


FIG. 70

as shown by the arrow, the arrangement shown in Fig. 70 will reverse the engine, the angle $C'O'E'$ being equal to the angle $COE$. With this arrangement no rocker-arm is employed between the eccentric-rod and the eccentric.

Figs. 71 and 72 show the arrangement when a rocker-arm

is employed. The angle $C'O'E'$ is equal to the angle $COE$, just as before, and the direction of rotation has been reversed, as shown by the arrows.

In both cases the angle through which the eccentric has been turned to effect the reversal is equal to 180°—twice the angle of advance.

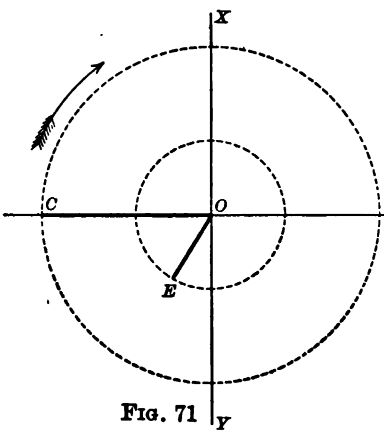



FIG. 71

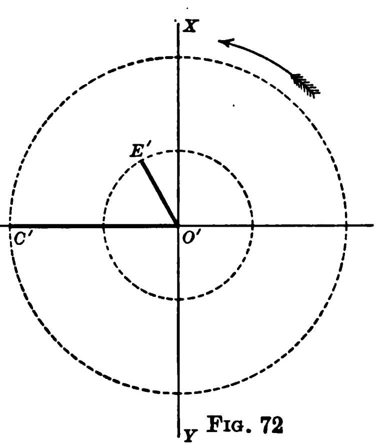



FIG. 72

There are numerous means by which the eccentric can be moved around the shaft; and they may be divided into two general classes:

*First Class.* Movable eccentrics with which the engine must be stopped in order to effect the reversal.

*Second Class.* Movable eccentrics with which the reversal can be effected while the engine is in motion.

FIRST CLASS.

This is by far the simpler class, and, for reasons which should be very obvious, it is but seldom employed, although it was used on early stationary and locomotive engines. Figs. 73 and 74 illustrate an eccentric of this type. The shaft carries a disk which is forged or cast on it, or fastened in

place. This disk is slotted in an arc of a circle about the center of the shaft $O$. The eccentric is loose on the shaft and is slotted in a similar manner. A bolt is used to fasten the two together. With the eccentric in one extreme position, such as the one shown in Fig. 73, the angle of advance is $XOD$. The arc of the slot is made of such a length that when the eccentric is slipped around so that it is bolted in its other extreme position, as shown in Fig. 74, the angle of advance is $YOD'$, equal to $XOD$ of Fig. 73. In both figures the path of the eccentric center is the dotted circle shown.

That this change in the angle of advance will result in a reversal of the engine should be understood from Chapter I. Figs. 75 and 76, the valve diagrams corresponding to Figs. 73 and 74 respectively, will aid in the comprehension of the fact that the events of the stroke occur at the same points in both cases. The angles of advance are laid off on opposite sides of the vertical in the two figures for the sake of emphasis, but, as explained in Chapter III, this is not necessary.

Fig. 77 shows another type of this class. The eccentric-slot is replaced by a pin which projects through the slot in the disk, and is secured in place by a jam-nut on the opposite side. The same results will be secured by this type as with the one shown in Figs. 73 and 74, and the diagrams 75 and 76 apply to it as well.

SECOND CLASS.

This is the more important class, possessing many points of advantage over the first class, and is the one which is employed on all or nearly all high-speed engines, where they are used as "shaft-governors."

The fundamental principle in this class is to have the eccentric slotted to clear the shaft, and, by moving the eccentric across the shaft, change the angle of advance as required.

Eccentrics of this class are of either one of two kinds:

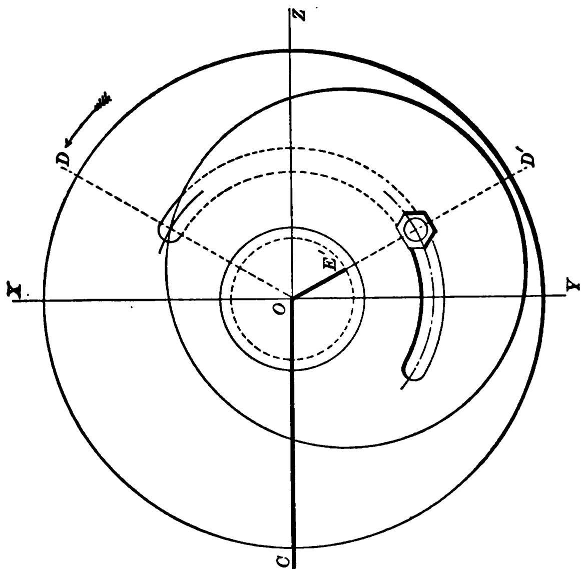



Fig. 74

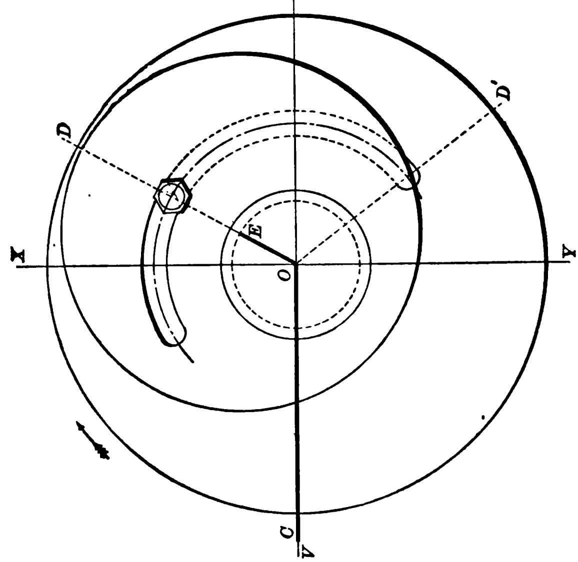



Fig. 73

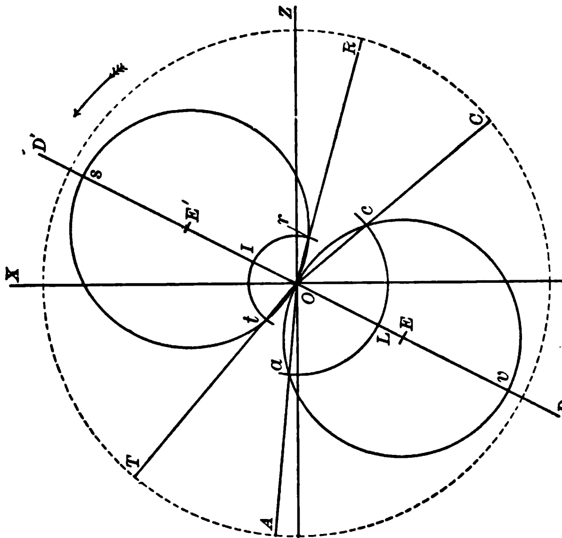



FIG. 76

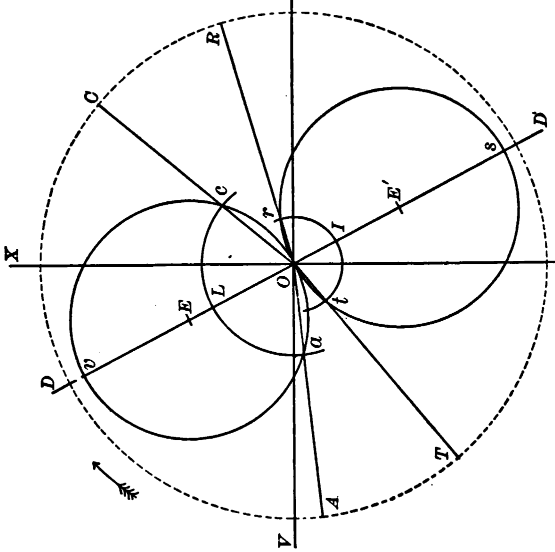



FIG. 75

1. "*Swinging*" *Eccentrics;* in which the eccentric is pivoted at some point which is secured to the shaft and rotates with it. In that case the eccentric swings across the shaft in the arc of a circle, and the slot is therefore curved.

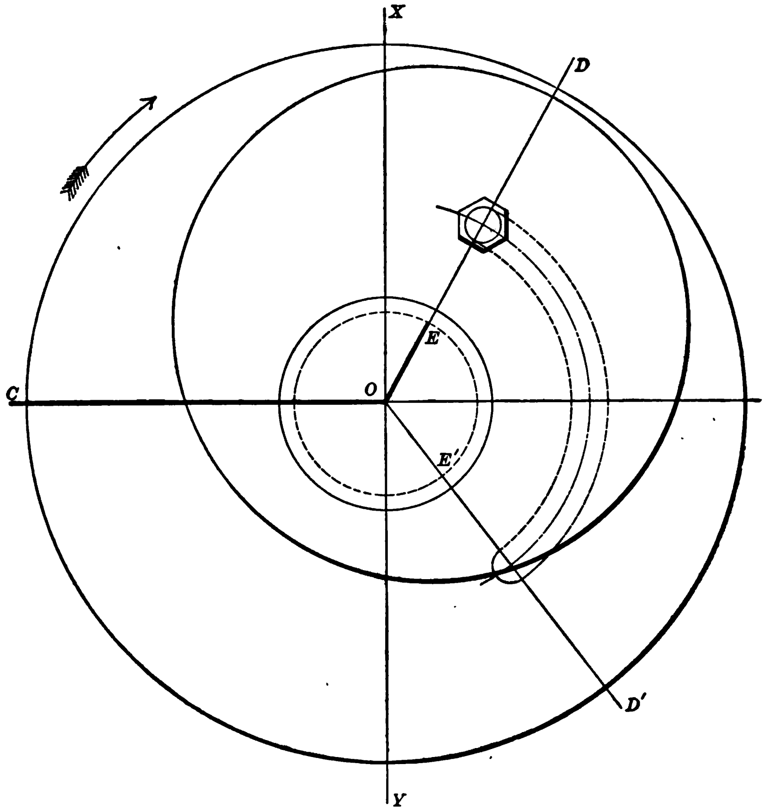



FIG. 77

2. "*Shifting*" *Eccentrics;* which move squarely across the shaft, and the slot is of course straight.

The first question to be considered is the effect of moving the eccentric across the shaft and holding it in any given position; the means by which this movement is effected being reserved for later discussion.

## SWINGING ECCENTRICS.

Figs. 78 and 79 show the two extreme positions of an eccentric of this class. It is pivoted at the point *P*, which is usually located on the arm of a small fly-wheel called the "governor-wheel" or "spider," which is securely keyed to the shaft. When in its upper position, Fig. 78, the angle between the crank and the eccentric is *COE*; and when in its lower position it is *C′O′E′*, Fig. 79. The shaded area in each

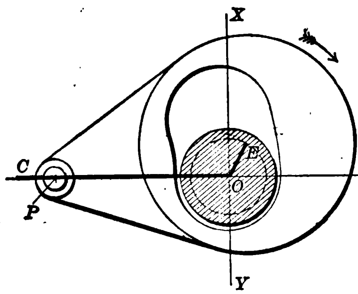



FIG. 78

FIG. 79

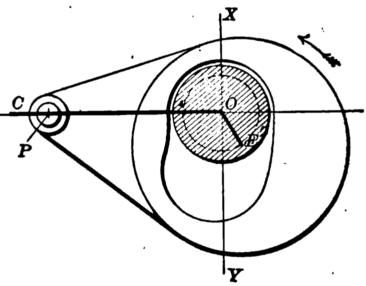



figure is the shaft, and the path of the eccentric center is shown by the dotted circle.

The manner of laying out an eccentric in this type is shown in Fig. 80. The point *P*, about which the eccentric is to swing, is generally fixed, and the center *O* of the shaft is also fixed. Then draw the line *VZ* through *O* and *C*, and through *O* draw *XY* perpendicular to *VZ*. The angle of advance having been determined, lay off *XOD* equal to that angle. Then describe the circle whose center is *O* and whose radius, *OE*, is equal to the eccentricity. This will cut the line *OD* at *E*, which is the center of the eccentric. Now about *P* as a center describe an arc passing through *E*. Again with *P* as a center describe an arc passing through the center of the shaft *O*. Now *E′*, which is where the eccentric

center must be in order to secure a reversal of the engine, will of course lie on the arc through $E$. Then the eccentric center must be shifted through the arc $EE'$ to secure the reversal. The next thing is to determine the length of slot required to allow this movement, and the smallest diameter of eccentric. Therefore from $e$, the point where the arc $EE'$ cuts the line $VZ$, lay off the arc $ee'$ equal to $EE'$. Draw $eP$, which cuts the arc drawn through $O$ at $O'$. Now the slot must be drawn in as shown. That is, the lower center is $O$, and the radius is just enough larger than the radius of the shaft to permit it to clear. The upper center is $O'$, and the radius is of course the same as the lower. The rest of the slot outline is made up of two arcs having $P$ as their center.

It is obvious that the center line of the crank must coincide with $VZ$, and the spider or containing-wheel of the governor must be keyed to the shaft so that this result will be obtained.

Now suppose the eccentric to have swung around so far that the center is at 1, Fig. 80. What is the result? The angular advance has been increased to $XO1$, and the eccentricity has been decreased to $O1$. (In order to have the eccentricity remain the same, it would be necessary to have the eccentric move around $O$ as a center.) Now, the effect of increasing the angular advance is to make the events of the stroke all occur earlier. Decreasing the travel makes the admission later, cut-off earlier, release earlier, and compression earlier, as shown in Table III. The combined effect of the two changes can best be understood by constructing the diagram. Fig. 81 shows the diagrams for five positions of the eccentric center in Fig. 80. Diagrams $E$ and $E'$ correspond to the extreme positions of the eccentric, or, as they are called, "full-gear forward" and "full-gear backward." Diagram 2 shows the steam distribution when the eccentric center is at 2, Fig. 80, midway between its two extremes; that is, when the eccentric is in "mid-gear." Diagrams 1 and 3 show the

results obtained with the eccentric center at 1 and 3 respectively, Fig. 80.

It will be noticed that the cut-off varies more than the admission or lead, because both changes—of angular advance

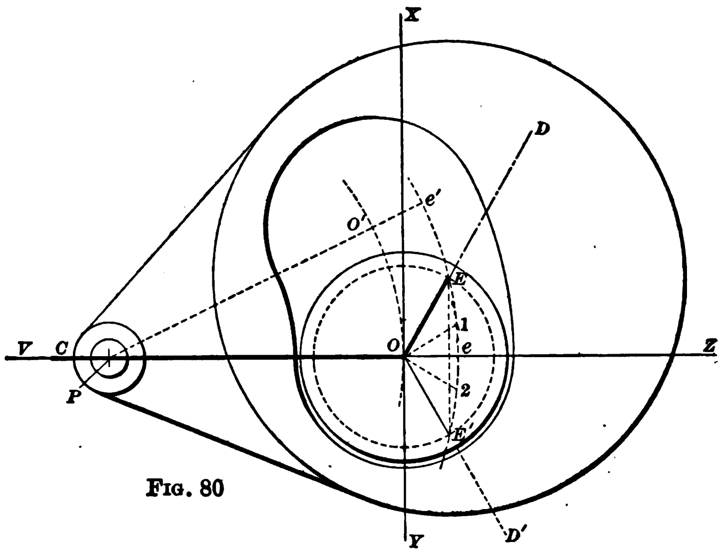



FIG. 80

and eccentricity—affect the cut-off in the same way, making it earlier; while the admission is made later by the decreased travel, and earlier by the increased angular advance. These two opposite changes in the lead may neutralize each other, but only when the center, $P$, about which the eccentric swings is removed to infinity—that is, when the arc through which it swings becomes a straight line, and the eccentric moves straight across the shaft. The valve diagrams show that the release varies less than the compression, with the swinging eccentric.

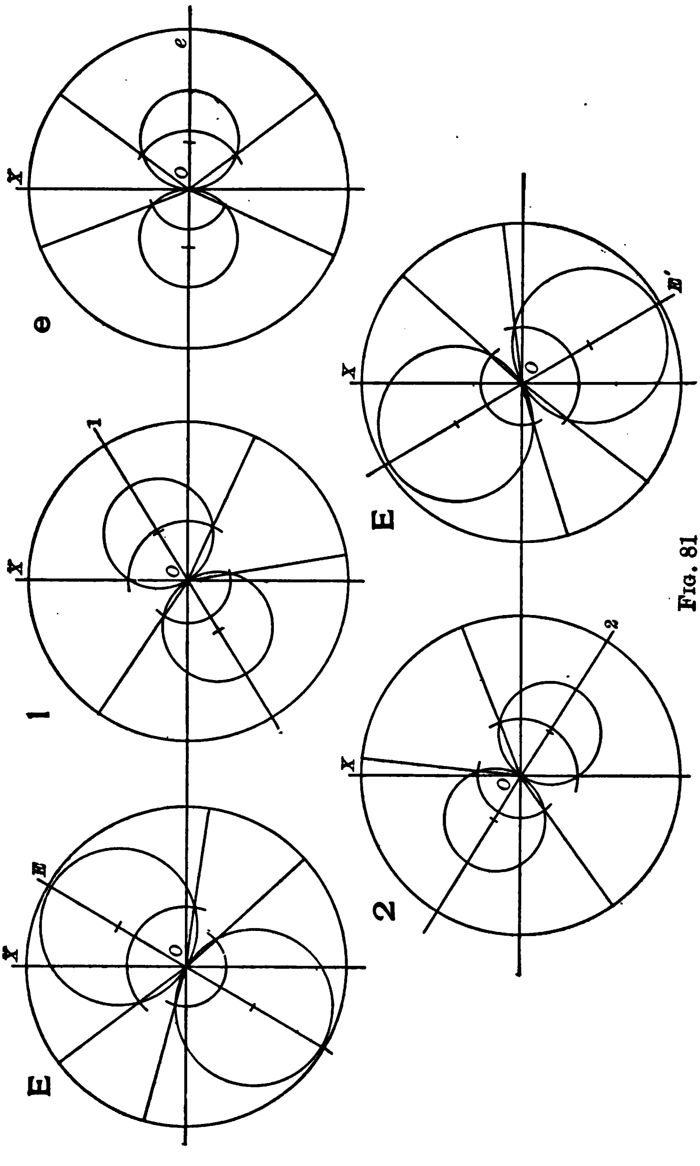



Fig. 81

## SHIFTING ECCENTRICS.

This is the type in which the lead is constant, as stated in the preceding paragraph. It differs from the swinging eccentric more in degree than in kind. Figs. 82 and 83 show one

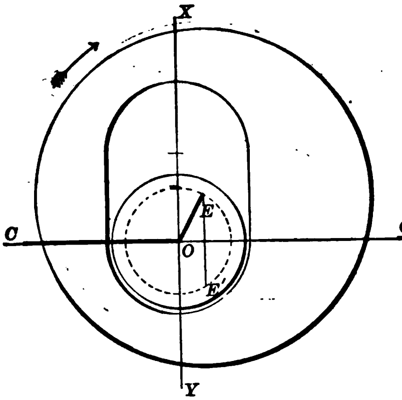



FIG. 82

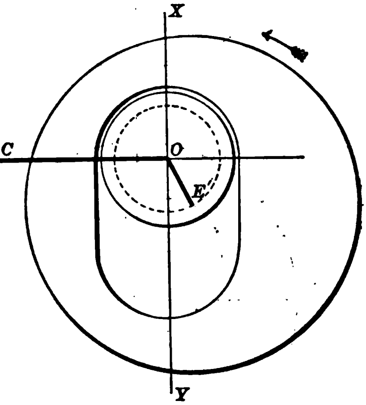



FIG. 83

of this kind in its two extreme positions. It will be noticed that the cut-off has a greater range of variation in this than in the swinging eccentric, because the valve travel has a greater range. This will be clearly understood by reference to Fig. 80, where the straight line from $E$ to $E'$ shows the path which would be followed by the center of a shifting eccentric designed to secure the same results as the swinging eccentric. The eccentricity being the distance from the center of the shaft to the center of the eccentric, the variation is evidently the greater with the shifting eccentric.

The next thing is to demonstrate that these variations in the events of the stroke can be made to serve a useful purpose.

Suppose an engine be running at full load; the eccentric

being fixed in place. Then suppose that a large part of the load is suddenly thrown off. If the steam-pressure remains the same, the engine will begin to speed up; and if enough load has been thrown off, the increase in speed may be sufficient to result in a bursting fly-wheel.

Now suppose that the engine is fitted with a movable eccentric, and that when the full load is on the eccentric is in full gear forward, giving the latest admission and latest cut-off. Again, suppose that the load is thrown off, just as before, but at the same time the eccentric is moved in toward the shaft. What will happen? Under the lighter load the engine will tend to speed up, but the earlier admission, due to the increased angular advance, will cushion the piston, tending to decrease the speed of the piston; and the cut-off will come earlier in the stroke, also tending to reduce the piston speed by shortening the length of time the moving force is applied. In other words, the period of admission, while remaining the same, is divided more nearly evenly between the two strokes.

Now, with the cut-off made earlier, the engine will not develop so much power; and the change may be just sufficient to adapt the engine to the lighter load. A light load at given speed requires less power than a heavy load at the same speed.

Figs. 84 to 87, inclusive, illustrate various forms of shaft-governors. The general principle of their construction is as follows:

A small wheel or spider is keyed on the shaft, as stated in the previous chapter. This wheel carries weights which are pivoted on the arms or ring, so that under the influence of centrifugal force they fly outward, this motion being opposed by springs. The weights are linked to the eccentric, so that a motion of the weights will cause a corresponding motion of the eccentric. Now, if the engine speed remains constant, the centrifugal force remains constant, and

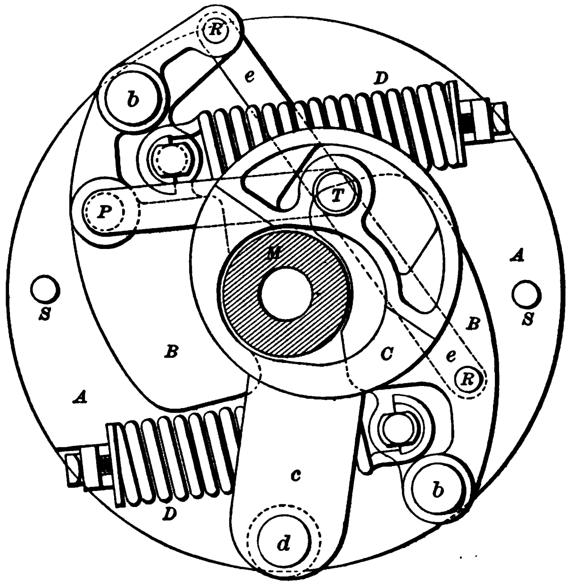



FIG. 84

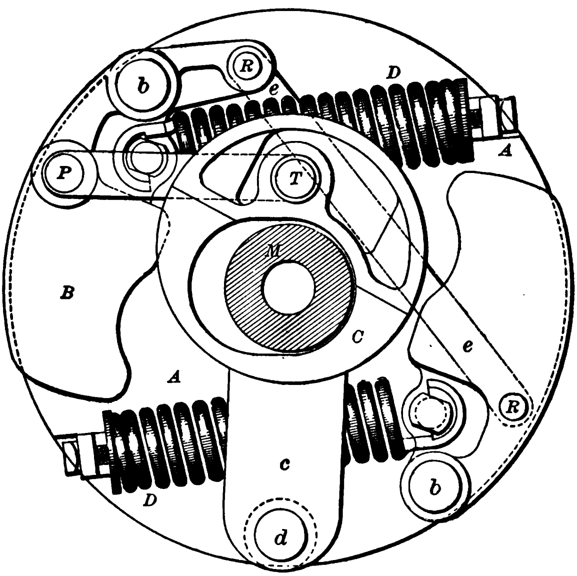



FIG. 85

the springs will be stretched a certain amount, and the eccentric will be held in one position. If, now, the load be lessened, the engine will speed up momentarily, increasing

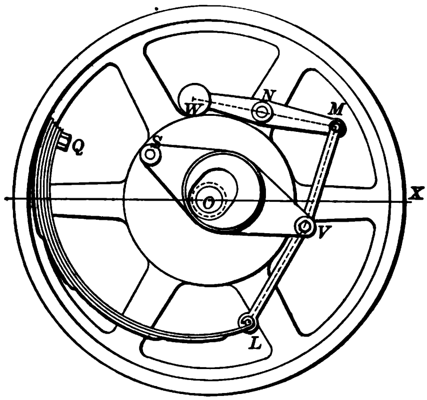



FIG. 86

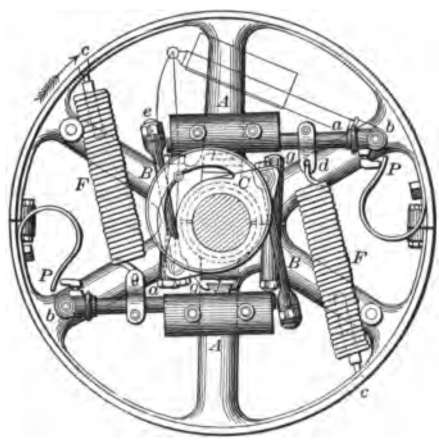



FIG. 87

the centrifugal force, thus throwing the weights outward and moving the eccentric across the shaft, shortening the cut-off and adapting the power to the load. If the load is increased,

the engine will slow down, decreasing the centrifugal force so that the springs will pull the eccentric back, lengthening the cut-off. In this way the events of the stroke are regulated so as to keep the speed of rotation sensibly constant at all loads.

Figs. 84 and 85 show two positions of the Westinghouse governor, Fig. 84 representing the governor when at rest or running below its normal speed; that is, at its latest cut-off. Fig. 85 shows the governor when in the position corresponding to its earliest cut-off; that is, when the engine has speeded up considerably above the desired point and the governor is acting to bring it down to the proper point.

The governor-weights *B* and *B* are pivoted to the wheel *A*, which revolves with the shaft *M* at *b* and *b*. The upper weight carries a link, *PT*, one end of which, *P*, is pivoted to the governor-weight, and the other end, *T*, is pivoted to the eccentric casting. The weights are connected by the link *ee*. The springs *D* and *D* are fastened to the weights and to the wheel or disk, as shown; thus resisting any tendency of the weights to fly outward. Under an increase of speed the centrifugal force increases, and the springs are extended, thus allowing the eccentric to move across the shaft.

Fig. 86 shows the Straight-Line governor, which is contained within the fly-wheel. The center of the shaft is at *O*. One of the fly-wheel arms, *N*, has a pivot on which the weight-lever, *WNM*, is hung. The end, *M*, of the lever is connected to the eccentric casting at *V* and to the spring at *L*, by the link *MVL*. The spring is secured to a boss, *O*, on the fly-wheel rim, as shown, and the eccentric is of the swinging type, being pivoted at *S*. The whole arrrangement occupies the position shown when the engine is at rest; it will be noticed that the eccentric is in its extreme position, thus securing the greatest cut-off, as required when the engine starts up. When the engine starts up the action of the centrifugal force will force the weight outward, thus moving the eccentric across the shaft.

Fig. 87 represents the Buckeye governor. When the engine is at rest the springs *FF* hold the weights *A A* against the inner stops. When the engine starts up the weights tend to fly outward; and when a certain rotative speed is reached they move away from the stops, thus stretching the springs. The eccentric being connected to the weights by the rods *BB*, it is moved around the shaft, thus causing an earlier cut-off. When a point of cut-off corresponding to the load has been established, the speed ceases to increase, and the spring pull and the centrifugal force will balance each other as long as there are no changes in the load on the engine. If the load is increased, or the pressure of the steam is reduced, the speed is reduced momentarily, and a later cut-off is established by the springs drawing the arms toward the shaft and changing the position of the eccentric, thus bringing the engine back to speed.

## CHAPTER XIII.

### SHAFT-GOVERNORS—ANALYSIS.

THE first principle of shaft-governors—that varying the events of the stroke properly will cause the engine to run at the same speed under all loads—having been established, the next point in order for consideration is an analysis of the action of the springs and weights whereby the movement of the eccentric is accomplished. The method pursued in this will be one drawn from a valuable monograph entitled "The Mechanics of the Shaft-Governor," by Prof. Barr, published in the *Sibley Journal of Engineering*, 1896.

First, reduce the springs and weights to the simplest form, shown in Fig. 88. Here there is but one weight, represented by the ball $W_0$, which slides on the radial rod $R$. The length of this rod is such that when the ball is in the extreme inner position its center coincides with the center of the containing-wheel or spider $G$. The movement of the ball is resisted by the spring $S$, which is fastened to the rim of the spider. The line of travel of the spring and ball is a diameter of the circle. In this investigation friction is neglected.

First, suppose that the ball is moved outward by any force from its central position until it occupies a position $W$. In that case the spring must be extended a certain amount, equal to the distance from $W_0$ to $W$, or the distance the ball is moved; and it will then exert a pull on the ball, tending to restore it to its original position. The amount of this inward pull is found as follows. "Spring strength" is the term used to designate the force in pounds required to extend or com-

press a spring one inch. Thus, a 20-pound spring means one which requires a pull of 20 pounds to extend it one inch; a 60-pound spring requires a pull of 60 pounds to accomplish the same result. The total amount of extension or compres-

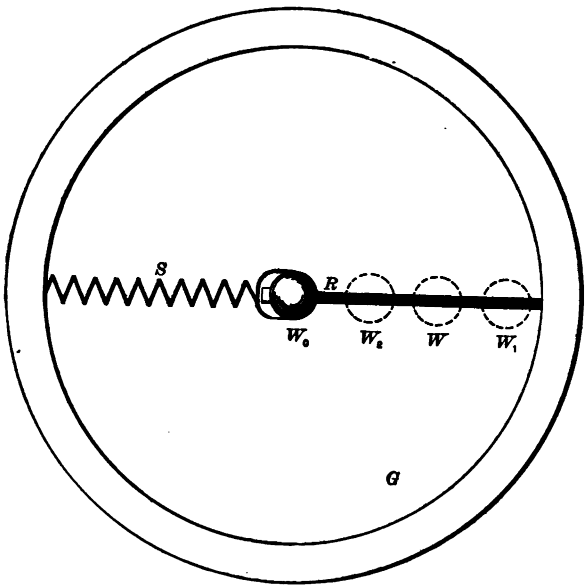



FIG. 88

sion is directly proportional to the force exerted upon the spring. Thus, a weight of 120 pounds resting on the 20-pound spring would compress it

$$120 \div 20 = 6 \text{ inches,}$$

or, if directly suspended from it, would lengthen it the same amount. The same weight would only lengthen or shorten a 60-pound spring

$$120 \div 60 = 2 \text{ inches.}$$

The weight or force required to produce any given change in the length is found by multiplying the spring strength by the

amount of the change in inches. For example, to lengthen a 30-pound spring $\frac{1}{2}$ inch would require a force of

$$30 \times \tfrac{1}{2} = 15 \text{ pounds.}$$

The force exerted by a spring is equal to the force exerted in compressing or extending it to the length it has. The 30-pound spring just mentioned would exert a force of 15 pounds if compressed or extended. "To every action there is an equal and opposite reaction."

Again, suppose the wheel to revolve with the ball at $W$. There will be a centrifugal force set up which will tend to throw the ball outward; this force will act radially, and as the ball is free to slide along the rod it may be considered that the centrifugal force acts along that line. The amount of this force depends upon the weight of the ball, the rotative speed of the wheel, and the distance from the ball to the center of the wheel. Expressed in a formula, it is

$$C = .0000284WN^2R, \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where $C =$ centrifugal force in pounds;
$W =$ weight of ball in pounds;
$N =$ revolutions per minute;
$R =$ radius, or distance from the center of the wheel to the center of the ball, in inches.

This may be stated in a rule as follows:

*To find the centrifugal force in pounds exerted by a weight: Multiply together the weight in pounds, the square of the number of revolutions per minute, and the radius, or distance in inches from the center about which the weight revolves to the center of gravity of the weight. Multiply this product by the constant* .0000284, *and the result is the centrifugal force.* (In the case of a ball, the center of gravity is the center of the ball.)

For example, suppose that the ball at $W$ weighs 20

pounds, is 20 inches from the center of the wheel, and that the latter is running at 150 revolutions per minute. The centrifugal force is 255.60 pounds, found by the above rule as follows: The square of the number of revolutions per minute is $150 \times 150 = 22{,}500$. Then the centrifugal force is

$$C = 20 \times 22500 \times 20 \times .0000284 = 255.60 \text{ pounds.}$$

Formula (1) may be transposed to give the value of $N$ when the other factors are known, giving

$$N = 187.7\sqrt{\frac{C}{WR}}. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This expressed in the form of a rule is as follows:

*To find the number of revolutions per minute which a ball of a known weight, at a known radius, must make to exert a given centrifugal force: Multiply the weight in pounds by the radius in inches, and divide the centrifugal force by this product. Then extract the square root of the quotient, and the figure is the number of revolutions per minute.*

For example, if a weight of 20 pounds is rotating about a center 20 inches distant, and exerting an outward pull of 255.60 pounds on the link connecting it with that center, it is making 150 revolutions per minute; because, applying formula (2) or its corresponding rule,

$$\begin{aligned} 20 \times 20 &= 400 \\ 255.60 \div 400 &= .639 \\ \sqrt{.639} &= .7994 \\ 187.7 \times .7994 &= 150.04 \end{aligned}$$

Now, if the ball when running at any given speed is held stationary, it is obvious that at that moment the centrifugal force of the ball and the spring pull must be equal to each other. That is, taking the 20-pound weight which was just figured to have 255.60 pounds of centrifugal force urging it

outward, the inward spring pull must be 255.60 pounds in order to hold the ball at $W_1$. It is obvious that this amount of spring pull could be secured by using a spring of any strength and stretching it the required amount; or the stretch of the spring could be assumed and the strength calculated. It is better to assume the spring strength, however, because this affects the closeness of regulation, as will be explained later.

Suppose, then, that the spring is a 40-pound. Then to hold the ball at $W$ the extension necessary is

$$255.60 \div 40 = 6.39 \text{ inches.}$$

This extension will not permit the ball, when the tension is relieved, to move in to the center of the containing-wheel; because the ball is 20 inches out from the center, so that, with the tension removed and the spring collapsed, the ball would still be

$$20 - 6.39 = 13.61 \text{ inches}$$

away from the center. The spring pull with the ball that distance out from the center would be zero, and therefore, to maintain a balance between the centrifugal force and the spring pull, the centrifugal force must be zero. This can only happen when the radius at which the weight revolves, or the rotative speed, is zero. But as the radius is 13.61, the rotative speed must be zero.

Now suppose that the ball moves inward to $W_2$, which is 4 inches from $W$, or 16 inches from the center. In that case the spring extension has been decreased to $6.39 - 4 = 2.39$ inches, and the spring pull to

$$2.39 \times 40 = 95.6 \text{ pounds.}$$

Then the speed at which the wheel must revolve to maintain the ball at $W_2$, the revolutions per minute, must be such

as to produce a centrifugal force of 95.6 pounds. Formula (2) or Rule II gives the value of $N$ as 113 07.

$$N = 187.7\sqrt{\frac{95.6}{20 \times 20}}$$
$$= 187.7\sqrt{.239}$$
$$= 113.67.$$

That is, the governor-ball will not move to $W_2$ until the speed has decreased to 113.67 revolutions per minute.

Next, find the effect when the ball is on the other side of $W$, say at $W_1$, which is 4 inches farther out than $W$, or 24 inches from the shaft center. The spring extension at that place is

$$6.39 + 4 = 10.39,$$

and the spring pull is

$$10.39 \times 40 = 415.6.$$

In order to have the centrifugal force equal to the spring pull, the revolutions per minute must increase to 175.52, because, from formula (2),

$$N = 187.7\sqrt{\frac{415.6}{20 \times 24}}$$
$$= 187.7\sqrt{.86583}$$
$$= 187.7 \times .9351$$
$$= 175.52.$$

That is, with the engine and governor-wheel running at 175.52 revolutions per minute the ball would be held in balance at $W_1$, 24 inches from the center of the shaft.

Then if $W_2$ and $W_1$ are the extreme inner and outer positions of the governor-ball, the extreme variation in speed is from 113.67 revolutions per minute to 175.52. If the ball in

moving from $W_2$ to $W_1$ moves the governor from its earliest to its latest cut-off, the engine will vary

$$175.52 - 113.67 = 61.85$$

revolutions per minute in passing from light to full load. The normal speed of the engine being taken at 150, the variation is

$$\frac{61.85}{150} = 41.23 \text{ per cent.}$$

Prof. Barr has devised a diagram which is of great service in showing clearly and at a glance the relation between the spring pull and centrifugal force. Fig. 89 shows it as applied to the case just discussed.

Draw the line $W_0W_1$, and let the distances on it from $W_0$, on any convenient scale, represent distances from the center of the shaft, or radii, at which the ball revolves. The scale adopted here is two small squares to the inch. Then let distances perpendicular to $W_0W_1$ represent the forces acting on the ball. For example, it was shown that with the ball at $W_1$, the spring pull, no matter what the speed, is 415.6 pounds. Then lay off $W_1S_1$ to represent this amount. The scale chosen in the figure is 20 pounds for one small square; and it will generally be found necessary to employ a much smaller scale for the forces than the distances in order to bring the drawing within reasonable limits. Next, lay off $W_2S_2$ equal to 95.6, which was found to be the spring pull with the ball at $W_2$, or 16 inches from the center. Join $S_1$ and $S_2$, and produce the line $S_1S_2$ until it meets $W_0W_1$ at $S_0$. It will be found that $S_0$ is 13.61 inches from $W_0$. This is as it should be; for it was shown that with the ball at that distance from the center the spring pull is zero. The line $S_0S_1$, representing the spring pull, is straight, because the pull varies directly as the extension of the spring. The spring pull at any point is then found by drawing a perpendicular from the point on $W_0W_1$, and measuring the length included between $W_0W_1$ and

the line of spring pull; thus with the ball at $W$, 20 inches from the center, the spring pull is $WS$, which is 255.6 pounds.

In other words, horizontal distances from $S_0$, as $S_0W_2$, $S_0W$, $S_0W_1$, represent the spring extension.

Now to consider the centrifugal force. It is desirable to have the engine run at a constant speed, and the centrifugal force will be considered on that basis. There must be some radius at which the spring pull and the centrifugal force at that speed will balance. For example, it was found that with

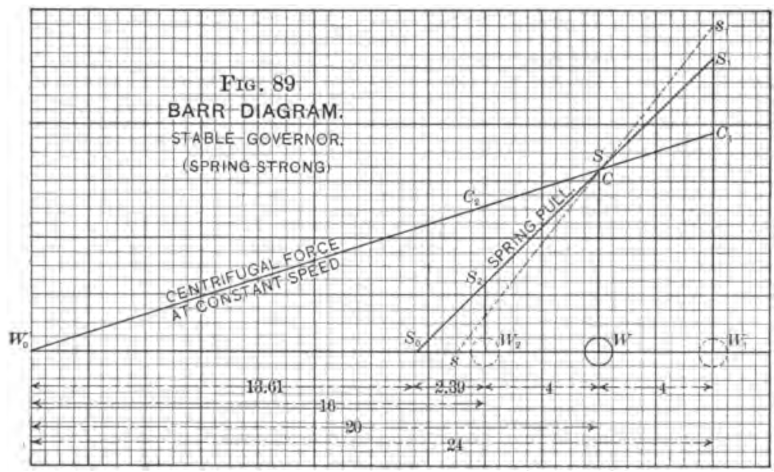



FIG. 89
BARR DIAGRAM.
STABLE GOVERNOR.
(SPRING STRONG)

the engine running at 150 revolutions, the spring pull and the centrifugal force balance with the ball at $W$, 20 inches out from the center. Now, with a constant speed, the centrifugal force varies directly as the radius, as shown by formula (1). With the ball at the center of the wheel—with the radius zero—the centrifugal force is zero. Therefore the lines of spring pull and centrifugal force at constant speed coincide at 20 inches from $W_0$, and $W_0C_1$ is the line of centrifugal force, found by drawing a straight line from $W_0$ through $C$. The centrifugal force at any other radius is found by drawing the

perpendicular from the end of the radius to the line of centrifugal force. Thus at $W_1$ the centrifugal force is $W_1C_1$, equal to 304.9 pounds, and with the ball at $W_2$ the centrifugal force is 202.5 pounds, measured by $W_2S_2$.

Now this diagram shows very clearly that if the speed be maintained constant, the centrifugal force and the spring pull will not be equal at different radii. They are equal at $W$ only; beyond that point the spring pull is greater, and below it the centrifugal force is the larger. This shows that the governor is possessed of a considerable amount of stability; that is, it is but little liable to derangement by outside forces, such as the drag of the valve. Suppose, for example, that the engine is running at 150 revolutions. Then the ball balances at $W$, and if the speed be maintained constant, a force equal to $S_1C_1$ must be applied to move the ball out to $W_1$ and balance it there. For, if the ball is at $W_1$, the centrifugal force urging it outward at 150 revolutions is $W_1C_1$; and the spring pull drawing it inwards is $W_1S_1$. The difference between these two is $C_1S_1$, equal to

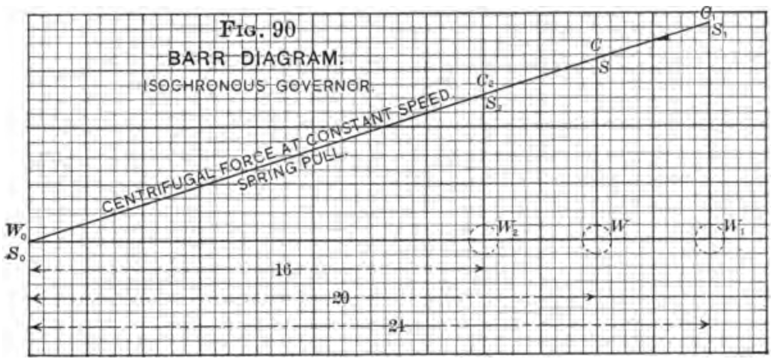
$$415.6 - 304.9 = 110.7 \text{ pounds,}$$

which is the force necessary to send the ball out to $W_1$ and hold it there. Of course, if a lesser force were applied, it would move the ball somewhat, but it would not force it out all the way.

The action of this type of governor, with a very strong spring, may be summed up as follows: It does not give very close regulation, the example chosen showing a variation of from 175 to 113 revolutions, but the stability, or resistance to external forces, is very great. This stability increases as the spring strength increases, which shows clearly on the diagram, the line *ss* being the spring pull for the stronger spring. This is because the point at which the spring pull is zero is at a greater distance from $W_0$, and the line of spring pull, passing through $S$, makes a greater angle with the line of centrifugal

force. Hence the distance between the two lines is greater at any given point; and as this distance measures the force required to displace the ball, the stability is greater.

Next, consider this form of governor having in a spring of such strength that when there is no spring extension the ball will be at the center of the spider. That is, with the ball out anywhere on the rod the spring extension is equal to the radius at which the ball revolves. The effect of this spring can be best understood by reference to Fig. 90, which is the




Barr diagram for this case. Take the same numerical example as before, a 20-pound ball running at 150 revolutions at a radius of 20 inches. The centrifugal force is 255.6 pounds as determined before, and the spring strength must be

$$255.6 \div 20 = 12.78 \text{ pounds}$$

to bring the weight to the center of the spider when the spring is collapsed, or has no tension on it. This figure is drawn on the same scale as the preceding one, and the line of centrifugal force at constant speed is the same as before; but the line of spring pull is decidedly different. At $W$ the spring pull is equal to the centrifugal force. At $W_0$ it is zero. Therefore the straight line joining these two points,

which represents the spring pull, coincides with the line of centrifugal force.

What does this show? It shows that the governor regulates very closely, for the spring pull is equal to the centrifugal force at all positions. With the ball out at $W_1$ the speed would be 150 revolutions, and with the ball at $W_2$ the speed would be the same. But in obtaining this closeness of regulation the stability has been sacrificed. The distance between the lines of spring pull and centrifugal force measures the resistance to deranging forces; and, as this distance is zero at all points, the slightest force would move the ball over its entire length of travel. The slightest increase of speed would send the ball out to its extreme outer position, and the slightest decrease would bring it in to the center of the wheel. A governor having this property is said to be isochronous.

Fig. 91 shows the Barr diagram for the remaining case of the elementary governor, that in which the spring is too weak for isochronous action. The same figures are used as before —weight of ball, 20 pounds, revolutions 150, spring pull and centrifugal force balancing at a 20-inch radius. Assume the spring strength as 8 pounds. Then, the centrifugal force with $W$ at 20 inches being 255.6 pounds, the spring extension must be

$$255.6 \div 8 = 31.95 \text{ inches.}$$

That is, the spring extension is

$$31.95 - 20 = 11.95 \text{ inches}$$

greater than the radius at which the ball revolves. This locates the point $S_0$ over at the left of $W_0$, as shown in the figure. Joining $S_0$ and $S$, which is located the same as in the previous cases, gives the line of spring pull. Joining $W_0$ and $S$ gives the line of centrifugal force at constant speed. This case is the reverse of the first one, shown in Fig. 89. Beyond

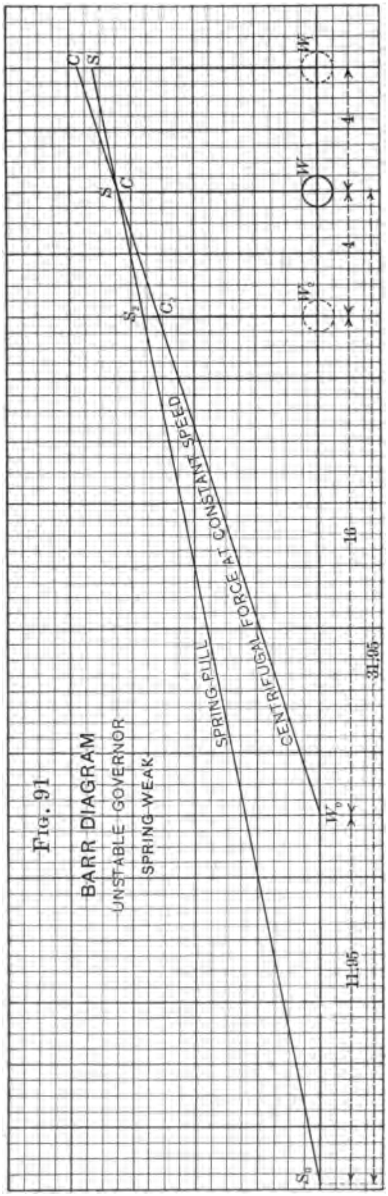


FIG. 91
BARR DIAGRAM
UNSTABLE GOVERNOR
SPRING WEAK
SPRING PULL
CENTRIFUGAL FORCE AT CONSTANT SPEED
C
S
W
4
16
31.95
11.95

$W$ the centrifugal force is greater than the spring pull, and below $W$ the spring pull is the greater of the two, thus showing that this form of governor is decidedly unstable. Suppose, for example, that this engine is running at its normal speed of 150 revolutions with the ball at $W$. Then suppose a part of the load to be suddenly thrown off. As a natural consequence the engine will speed up, and the increased centrifugal force will throw the ball out; and as the centrifugal force increases faster than the spring pull, the ball will go clear out to its extreme position, thereby making the cut-off much earlier than is necessary for the change in the load. This early cut-off of course results in a reduction of speed. If the outer limit is $W_1$, where the spring extension is

$$(31.95 + 4) = 35.95 \text{ inches},$$

and the spring pull is

$$35.95 \times 80 = 284.60 \text{ pounds},$$

the ball will balance there until the speed is reduced below the point where the centrifugal force equals 284.6 pounds, or, from formula (2),

$$\begin{aligned} N &= 187.7\sqrt{\frac{284.6}{20 \times 35.95}} \\ &= 187.7\sqrt{.3964} \\ &= 187.7 \times .629 \\ &= 118.06 \text{ revolutions per minute.} \end{aligned}$$

As soon as the speed is reduced below this point the ball begins to move inward; and as the centrifugal force decreases faster than the spring pull, it will continue to move in until it reaches its inner limit. This will result in a later cut-off than is necessary to bring the engine back to speed, and the result is that the speed will grow greater than is necessary to hold

the ball at $W_1$. If this inner position is, as before, at $W_2$, which is 16 inches from the center of the wheel, the spring extension at that point will be

$$16 + 11.95 = 27.95 \text{ inches,}$$

and the spring pull will be

$$27.98 \times 8 = 223.60.$$

The revolutions required to produce this amount of centrifugal force with the ball out 16 inches are

$$\begin{aligned} N &= 187.7\sqrt{\frac{223.6}{20 \times 16}} \\ &= 187.7\sqrt{.69875} \\ &= 187.7 \times .836 \\ &= 156.92. \end{aligned}$$

That is, the engine must speed up to about 157 revolutions before the ball will move out again. When it does start, it will go out to the outer end again, and it will keep on "hunting" or "racing" up and down, the speed meanwhile varying from 118 to 157 revolutions, or

$$\frac{(157 - 118)}{150} = 26 \text{ per cent.}$$

The same argument holds true for any other deranging force, such as the weight of the valve pulling on the valve-rod, so that this form of governor is evidently very unstable.

These three cases—stability, isochronism, and unstability—cover the ground completely. The first two qualities are greatly to be desired, but it will be seen at once from the preceding text that they cannot be obtained at the same time. Stability can be secured easily enough by using a strong spring, but this renders isochronism out of the ques-

tion. In the real governor, friction in the moving parts renders the governor stable, but at the same time it destroys the desired isochronism. Good results have been secured by reducing the external forces acting on the governor to a minimum and then making the governor as nearly isochronous as possible. It was shown that a perfectly isochronous governor is a possibility when the spring pull and centrifugal force are considered to be the only forces acting on the weight. But when gravity is considered it will be found that the perfectly frictionless governor will race between its outer and inner limits under a steady load. This is due to the fact that when the ball is above the shaft, gravity tends to draw it in, and when the ball is below the shaft, gravity tends to draw it out, or away from the center. The average cut-off obtained by this action would be the one suited to the load, but would be alternately too early and too late. This could, of course, be obviated by making the friction of the governor enough to prevent the racing, but this is a poor plan, as the desired isochronism is thereby rendered impossible, because, when the ball moves outward to compensate for a reduction of the load or for a reduction of the steam-pressure, the centrifugal force must overcome the friction in addition to the spring pull. The best method of overcoming the gravity distortion is to balance the governor; that is, to employ two weights, or their equivalent, one on each side, the result being that the gravity effects on the two will be opposed to each other and will therefore be negligible.

The same argument applies to the disturbing force of the valve; it could be compensated for by friction of the governor, but it is not desirable to so arrange it, on account of the destruction of isochronism. With an ordinary slide-valve this pull would be very great, because the steam of a pressure equal to or but little less than the boiler-pressure bears directly upon the valve, forcing it against the seat, while the pressure underneath the valve acting upward against this is

only that of the exhaust steam which is expanded to a considerably lower pressure. This unbalanced downward pressure increases greatly the effort required to move the valve. When this difference of pressure is obviated or greatly reduced, the valve is said to be balanced.

The simplest form of balanced valve is the piston-

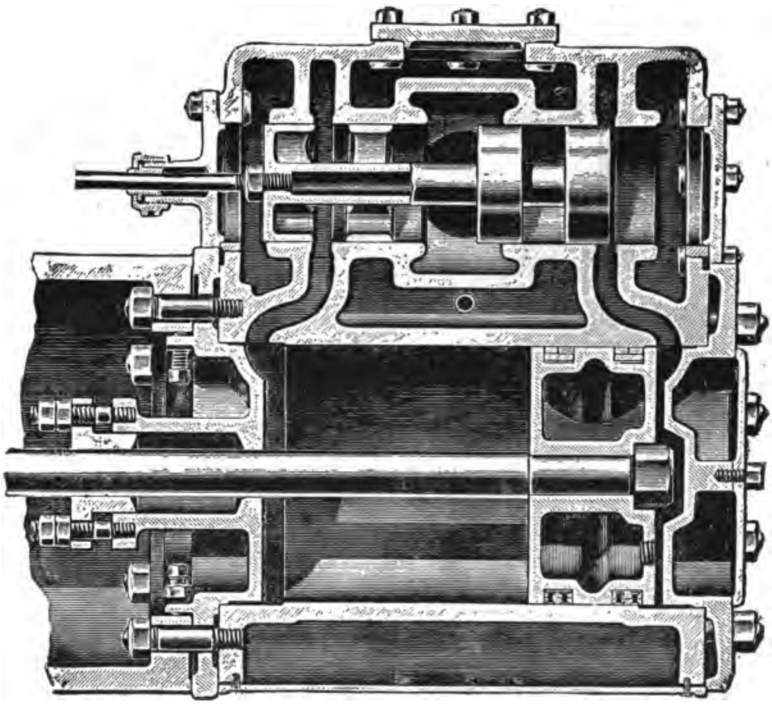


FIG. 92.

valve, shown in section in Fig. 92. The valve is cylindrical, the shape being similar to that which would be obtained by rotating a plain D valve about its valve-stem. Consequently the valve-seat must be cylindrical to fit the valve-seat. The valve may be arranged with double pistons, such as in Fig. 92, or the pistons may be single, as in Fig. 93. By referring to either figure, it will be seen that the steam-pressure must

be equal on the two end faces of the valve; and as the valve is in contact with the seat throughout its entire length, there is no possibility of any other pressure than friction getting at the valve sidewise. The objections to this form of valve are leakage and wear. The piston cannot fit tight in its bore, because unequal expansion would cause the piston to bind in the bore when the latter is cold and the pistons hot. This necessary difference in size would result in leakage if not

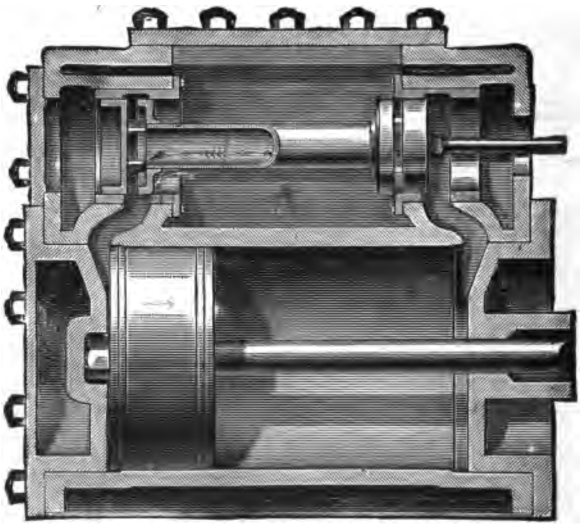


FIG. 93.

guarded against. Spring rings are used to effect the steam-tight joint, and these rings cause friction and wear. Sometimes no rings are used, but in that case great care must be taken to keep the temperature of the piston and the bore the same. This is done by steam-jackets, as shown in Fig. 93.

Another system of balanced valves employs pressure-plates. Here a flat plate is used which is secured in the steam-chest, and which receives the unbalanced steam-pressure, while the valve slide sunder the plate. It will be seen that the principle is practically the same as that of the piston-

valve, with the difference that with the pressure-plates the valve can be made flat and steam-tight at the top and bottom only, where it touches the pressure-plate and seat.

There are three classes of pressure-plates—fixed, adjustable, and flexible. Fig. 94 shows one of the fixed type. Here the plate is bolted to the bottom of the steam-chest, and the length of the plate is such that the valve never projects

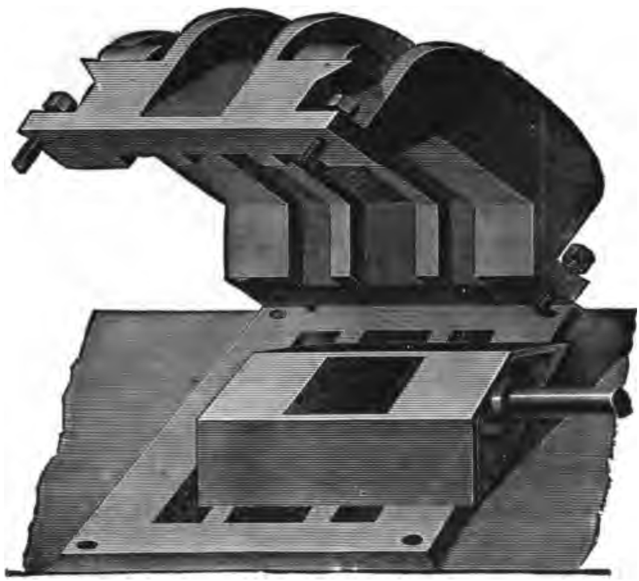


FIG. 94.

beyond it. The exposed ends of the valve, being of equal area, balance each other, and the pressure under the valve through the ports is balanced by recesses of equal area under the hood and over the valve.

In some cases a spring ring or similar arrangement is fitted to the top of the valve in such a way as to make a continuous contact between the plate and the valve, thus preventing any steam from acting on the top of the valve. Fig. 95 shows such a valve, where the rings *pp* are inserted in the top or

back of the valve and are pressed upward against the plate *A* by means of springs. The space between the plate and the back of the valve is open to the exhaust through the opening *h*, as shown. This balances nearly all the top of the valve, and the pressure on the remaining portion is sufficient to prevent leakage.

An example of the second type is shown in Fig. 96. In this design the pressure-plate is supported on an inclined plane, the plate being made to slope at the same angle, as shown by the dotted line in the figure. By means of the adjustable handle *E*, the movable plate can be adjusted to

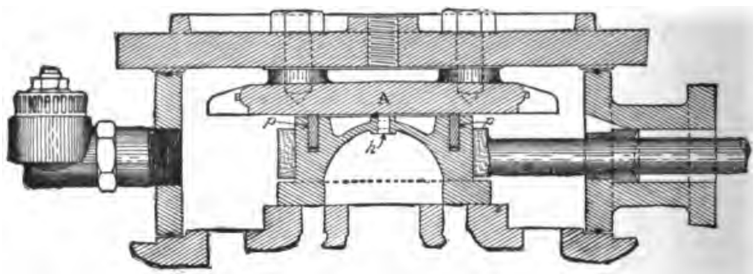



FIG. 95.

secure any desired amount of pressure on the back of the valve. The valve also illustrates another type of double-ported valve.

The third system, that of flexible pressure-plates, consists of a flexible plate of steel or other elastic metal, which is so arranged as to allow the pressure upon it to force it down upon the valve, but only with force enough to prevent leakage between the valve and the plate. An example of this type is shown in Fig. 97.

By the use of such valves as these the friction is greatly reduced, and the pull on the governor becomes a minimum.

Another disturbing element in the action of a governor is the inertia of the valve. This comes into play at each end of the valve travel, when it is necessary to reverse the direction

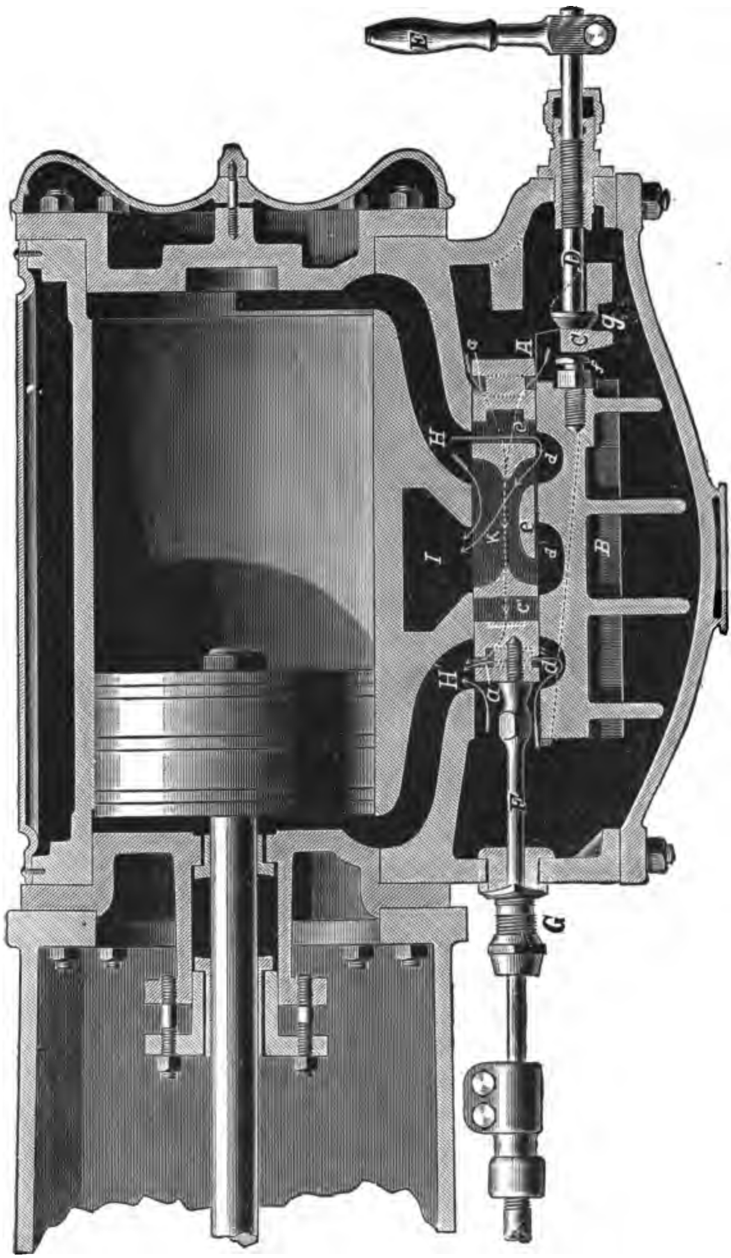



FIG. 96.

of motion of the valve, thus giving a sudden pull. One method of obviating or compensating this deranging force consists of employing a dash-pot. This consists of a piston fitting loosely in a cylinder containing oil. If the piston is moved slowly and steadily, the oil offers little or no resistance to the motion, as it will readily slip past the circumference of the piston. But if the piston be moved suddenly, the oil will be unable to flow by quickly enough to permit the piston to move rapidly, and will therefore bank up and impede the progress of the piston. A piston-rod passes through a

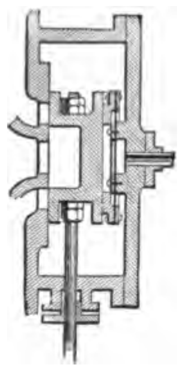


FIG. 97.

stuffing-box on the dash-pot, and is attached to the governor-weight, the dash-pot itself being fastened to and revolving with the governor-wheel or spider. This dash-pot corrects the distortion due to the inertia of the valve, and in no way impairs the action of the governor. Another result is obtained by the employment of this contrivance. When the weight starts to change its position under the influence of a change of speed it starts with a velocity which, owing to its inertia, would carry it beyond the proper place were it not for the retarding influence of the dash-pot. The dash-pot then serves as a preventive of too sudden a motion in either direction.

The last-mentioned deranging force or disturbing element —the inertia of the governor-weight—may be turned from a hindrance to a help, making it perform the functions of a dash-pot. This is explained as follows by Mr. E. J. Armstrong, in a paper read before the American Society of Mechanical Engineers, and forming a part of Vol. XI. of the *Transactions:* When any governor is engaged in its task of controlling the engine, the weight travels at a variable rate of speed, resulting from its revolving in a circle of variable size. This change in velocity is often quite considerable, depending of course upon the amount of radial movement and the rotative speed. To take an example from the Straight-Line engine: a fly-weight is 16⅞ inches from the center of the shaft when in, and 20¼ inches when out, making, at 220 revolutions per minute, a difference of 194 feet 4 inches per minute. Whenever the fly-weight takes a new position it must change its speed—must move either faster or slower, be accelerated or retarded—and must absorb or give out power somewhere. This resistance to a change in velocity acts at right angles to a radial line drawn through the center of gravity of the weight, and if the weight were pivoted so as to move radially, as in Fig. 98, the only result would be to increase the pressure on the pivot. If the fly-weight were so pivoted as to move at an angle to a radial line, as in Fig. 99, so that in its outward movement it goes toward the way the wheel rotates, then the outward movement of the fly-weight will be opposed, to some extent, by this resistance to acceleration, depending on the angle which the line of movement forms to a radial line—or, to put it in another way—upon the length of the lever-arm *AB*, *CA* being the line of resistance, and *B* the fly-weight pivot. When the weight moves toward the shaft, the action is the same. It has to part with some of its momentum, and so hangs back, as in its outward movement, thus making, similarly to a dash-pot, a resistance to movement in both directions, which can only be overcome

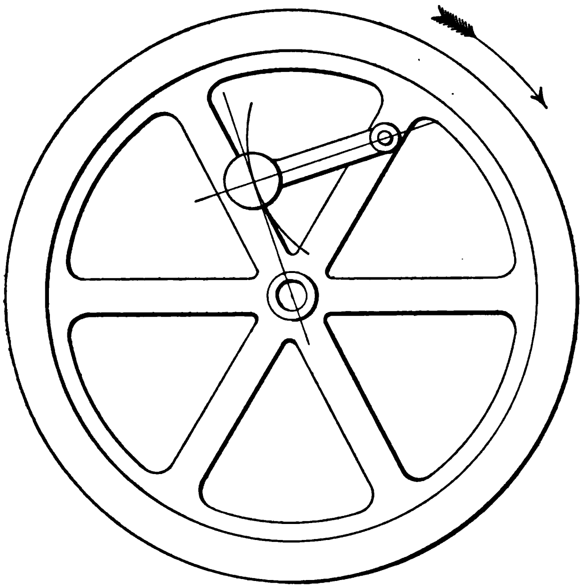


FIG. 98

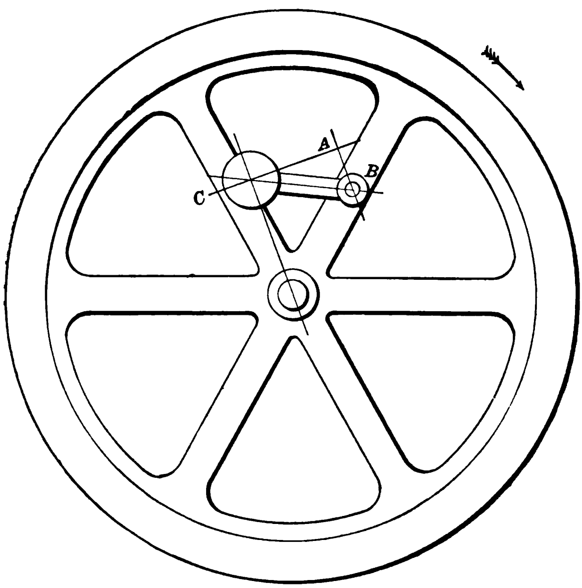



FIG. 99

quickly by a great force, or by a small one moving slowly; this resistance increases with the velocity of the weight, which is all that is accomplished by the ordinary dash-pot.

To return to the analysis of the governor proper. Fig. 100 represents a slight modification of the elementary type. The change consists in putting a stop, *T*, on the radial rod, for the purpose of preventing the ball from traveling in to the center of the wheel. With the wheel at rest it is possible

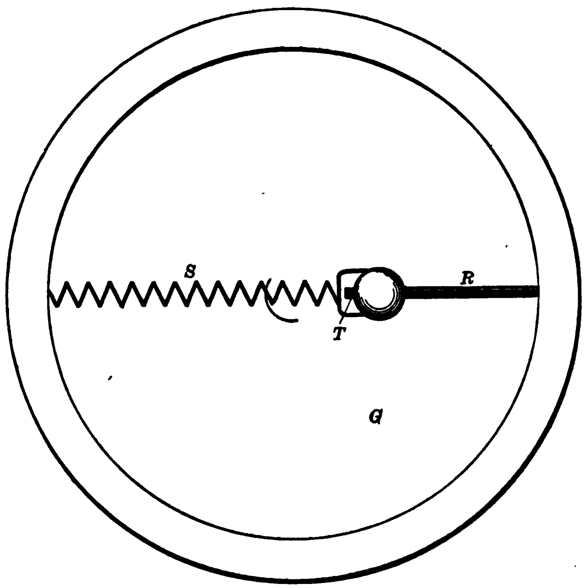



FIG. 100

to maintain the ball in the position shown in the figure by either one of two methods: First, by the employment of a spring of any strength, but of such length that it will hold the ball in that position without any extension—that is, the governor will be of the stable type, such as explained before, and whose action is illustrated in Fig. 89; and, second, by shortening such a spring, and then extending it by the amount

that it is shortened, thus getting an inward pull on the ball. This pull is called the initial tension, because it is the tension existing in the spring while the wheel is at rest, or before any centrifugal force is developed to extend the spring. The amount of the initial tension of the spring may be expressed in two ways—the amount, in inches, that the spring is shortened, or the pull, in pounds, on the ball. Suppose for example that a 10-inch spring of 40 pounds strength will hold the ball at the stop without any extension. If, then, this spring is shortened to 7 inches and then stretched out to cover the 10, the pull required will be

$$(10 - 7)40 = 120 \text{ pounds},$$

and if the end is fastened to the ball, it will exert that pull on it. The amount of the initial tension in pounds is therefore equal to the amount that the spring is extended when the wheel is at rest, and the ball against the stop, multiplied by the spring strength. If the initial tension is expressed by the inches that the spring is shortened, the spring strength must also be specified in order to render it exact. In the practical governors it is usual to supply some device, such as a screw-thread, by which the extension of the spring can be altered at will, and any desired amount of initial tension secured.

If it is desired to produce an isochronous governor, it is very evident that the initial tension must be such that if the stop on the radial rod were removed, the ball would move inward to the center of the retaining wheel.

If the initial tension is less than that required to bring the ball in to the center of the wheel with the stop removed, the result will be the same as if a strong spring were employed; that is, the governor will be stable.

If the initial tension is too great for isochronism—that is, if the ball would, on removal of the stop, go in beyond the center—the result obtained would be the same as if a weak spring were employed, as discussed and shown in Fig. 91.

This initial tension produces a slight variation in the operation of the governor. It will be remembered that with the elementary isochronous governor the weight starts out as soon as the engine is started; but with initial tension the engine must continue to speed up until it is going so fast that the centrifugal force of the ball exceeds the initial tension. For example, if the initial tension is 120 pounds, and if the ball, weighing 20 pounds, is held by the top at the point 6 inches from the center of the wheel, the engine will speed up until the centrifugal force is greater than 120 pounds; that is, from from the rule on page 127, until the revolutions per minute are greater than

$$
\begin{aligned}
N &= 187.7\sqrt{\frac{120}{20 \times 6}} \\
&= 187.7\sqrt{1} \\
&= 187.7.
\end{aligned}
$$

The previous argument being mastered, it is next in order to discuss the simple type of shaft-governor shown in Fig. 101. Here the weight $C$ is secured to the lever $AC$, the latter being pivoted at $A$ on an arm of the containing-wheel. The spring is fastened to the lever at $B$, and to the rim of the containing-wheel at $D$. Now, when the wheel revolves, a certain centrifugal force is induced which tends to throw the weight out, that is, to turn the lever about $A$. This is resisted by the spring as before, but with a decided modification. The spring pull and the centrifugal force are not directly opposed to each other, but act on the lever $AC$ at different distances from $A$. The distance $AB$ is the spring leverage, and the distance $AC$ is the weight leverage. It is at once apparent that in order to secure a balance between the opposing forces at any given speed, it must be true that spring pull $\times$ spring leverage $=$ centrifugal force $\times$ weight leverage. Or, putting this in symbols,

$$S \times L = C \times L',$$

where $S$ is the spring pull;
$L$ is the spring leverage;
$C$ is the centrifugal force;
$L'$ is the weight leverage.

Another difference between this and the elementary type is occasioned by the fact that the spring and weight are not

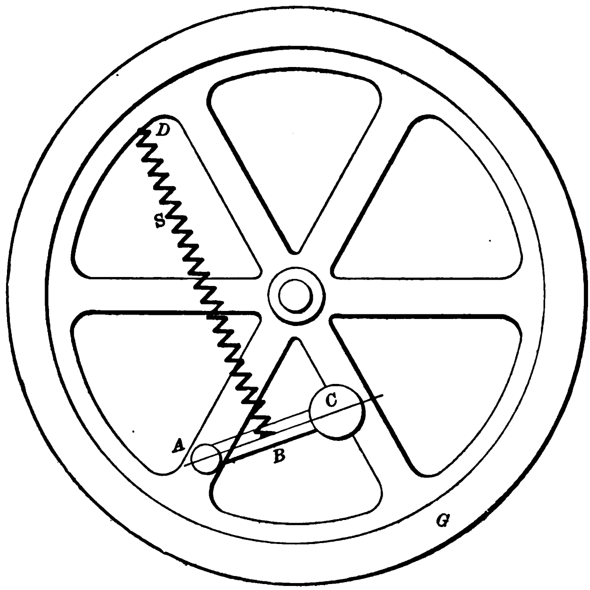



FIG. 101

directly opposed to each other; that is, the spring extension is not equal to the weight displacement. The spring extension is directly proportional to its leverage, and is found by multiplying the displacement of the weight by the spring leverage, and dividing by the weight leverage. For example, suppose the weight leverage to be 8 inches, while the spring leverage is but 4 inches, and that the weight has moved out

6 inches from its inner position under the influence of centrifugal force. Then the spring extension is equal to

$$\frac{6 \times 4}{8} = 3 \text{ inches.}$$

That is, if the spring has half the leverage of the weight, it has half the extension, and so on. It must be borne in mind that this extension is from the inner position of the weight. If a stop is used and initial tension is supplied, the total spring extension is equal to that above found, and which may be called centrifugal extension, increased by the initial tension in inches.

This form of governor possesses certain obvious advantages. When it is desired to secure a greater spring pull, it is not necessary to substitute a stronger spring. The same effect may be secured by fastening the spring at a greater distance from $A$, thereby securing a greater spring leverage, and at the same time increasing any existing initial tension, or, if none exist, adding some.

The practical governors have weights of various shapes, but the principle remains the same. The weight is supposed to be concentrated at its center of gravity, and the weight leverage is the distance from this center of gravity to the pivot, irrespective of the form of the connecting link. Of course the weight of the lever must be taken into consideration in finding the center of gravity. The radius at which the centrifugal force is found is the distance from the center of gravity to the center of the containing-wheel.

The foregoing text leads to the following general rules, which are applicable to all governors built on the above principle; and this type of governor is by far the most common.

### TO INCREASE THE SPEED.

It may be desirable to increase the speed at which the engine is to run. In that case, increase the spring tension, observing the following limitations: If the increase of spring tension is sufficient to produce isochronism, the governor will race as shown before. The increase must be kept below this limit, and as the governor is usually set near this limit, the above adjustment is applicable only to small changes of speed. For larger changes the governor-weight may be decreased. This will produce an increase of speed, because the centrifugal force will be reduced and it will require a greater number of revolutions per minute to develop sufficient force to move the weight across its extreme travel. Or the increase may be secured by moving the weight in toward the pivot about which it revolves. This lessens the weight leverage, and consequently a greater number of revolutions per minute are required to develop the same centrifugal force as before. An increase of the spring leverage will produce an increase of speed, because, the spring pull being thereby increased, a greater centrifugal force is required to stretch it, and this can only be secured by an increase in the number of revolutions per minute.

### TO DECREASE THE SPEED.

The adjustments here are necessarily the opposite of those cited above. For small changes, reduce the initial tension. For larger changes, the weight may be increased; or the weight may be moved out from the pivot, increasing the weight leverage; or the spring leverage may be reduced.

### TO REDUCE THE VARIATION OF SPEED.

That is, to produce isochronism. Increase the spring tension or diminish the spring leverage.

All of these changes are subject to one limitation—that

of performance. It is impossible to set a governor to produce exact results by theory alone. The adjustment must be made as nearly right as may be, and then the engine must be run. If the engine runs as calculated, all well and good. If not, the adjustment must be carried out on the lines indicated above until the engine runs smoothly and regulates closely.

## TABLE I.

VALUES OF $OO'$, FIG. 20, FOR VARIOUS RATIOS OF CONNECTING-ROD TO CRANK.

| Ratio of Connect'g-rod to Crank. | $OO'$ = Crank. | | Ratio of Connecting-rod to Crank. | $OO'$ = Crank. | | Ratio of Connect'g-rod to Crank. | $OO'$ = Crank. | |
|---|---|---|---|---|---|---|---|---|
| | Divided by | Multiplied by | | Divided by | Multiplied by | | Divided by | Multiplied by |
| 1 | 4 | .2500 | 4 | 16 | .0625 | 7 | 28 | .0356 |
| $1\frac{1}{4}$ | 5 | .2000 | $4\frac{1}{4}$ | 17 | .0588 | $7\frac{1}{4}$ | 29 | .0345 |
| $1\frac{1}{2}$ | 6 | .1667 | $4\frac{1}{2}$ | 18 | .0556 | $7\frac{1}{2}$ | 30 | .0333 |
| $1\frac{3}{4}$ | 7 | .1429 | $4\frac{3}{4}$ | 19 | .0526 | $7\frac{3}{4}$ | 31 | .0323 |
| 2 | 8 | .1250 | 5 | 20 | .0500 | 8 | 32 | .0313 |
| $2\frac{1}{4}$ | 9 | .1119 | $5\frac{1}{4}$ | 21 | .0476 | $8\frac{1}{4}$ | 33 | .0303 |
| $2\frac{1}{2}$ | 10 | .1000 | $5\frac{1}{2}$ | 22 | .0455 | $8\frac{1}{2}$ | 34 | .0294 |
| $2\frac{3}{4}$ | 11 | .0909 | $5\frac{3}{4}$ | 23 | .0435 | $8\frac{3}{4}$ | 35 | .0286 |
| 3 | 12 | .0833 | 6 | 24 | .0417 | 9 | 36 | .0278 |
| $3\frac{1}{4}$ | 13 | .0769 | $6\frac{1}{4}$ | 25 | .0400 | $9\frac{1}{4}$ | 37 | .0270 |
| $3\frac{1}{2}$ | 14 | .0714 | $6\frac{1}{2}$ | 26 | .0385 | $9\frac{1}{2}$ | 38 | .0263 |
| $3\frac{3}{4}$ | 15 | .0667 | $6\frac{3}{4}$ | 27 | .0370 | $9\frac{3}{4}$ | 39 | .0256 |

## TABLE II.

**DECIMAL EQUIVALENTS OF FRACTIONS OF AN INCH.**

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 1/64 | .015625 | 17/64 | .265625 | 33/64 | .515625 | 49/64 | .765625 |
| 1/32 | .03135 | 9/32 | .28125 | 17/32 | .53125 | 25/32 | .78125 |
| 3/64 | .046875 | 19/64 | .296875 | 35/64 | .546875 | 51/64 | .796875 |
| **1/16** | .0625 | **5/16** | .3125 | **9/16** | .5625 | **13/16** | .8125 |
| 5/64 | .078125 | 21/64 | .328125 | 37/64 | .578125 | 53/64 | .828125 |
| 3/32 | .09375 | 17/32 | .34375 | 19/32 | .59375 | 27/32 | .84375 |
| 7/64 | .109375 | 23/64 | .359375 | 39/64 | .609375 | 55/64 | .859375 |
| **1/8** | .125 | **3/8** | 375 | **5/8** | .625 | **7/8** | .875 |
| 9/64 | .140625 | 25/64 | .390625 | 41/64 | .640625 | 57/64 | .890625 |
| 5/32 | .15625 | 13/32 | .40625 | 21/32 | .65625 | 29/32 | .90625 |
| 11/64 | .171875 | 27/64 | .421875 | 43/64 | .671875 | 59/64 | .921875 |
| **3/16** | .1875 | **7/16** | .4375 | **11/16** | .6875 | **15/16** | .9375 |
| 13/64 | .203125 | 29/64 | .453125 | 45/64 | .703125 | 61/64 | .953125 |
| 7/32 | .21875 | 15/32 | .46875 | 23/32 | .71875 | 31/32 | .96875 |
| 15/64 | .234375 | 31/64 | .484375 | 47/64 | .734375 | 63/64 | .984375 |
| **1/4** | .25 | **1/2** | .5 | **3/4** | .75 | **1** | .1 |

## TABLE III.

EFFECT OF CHANGING OUTSIDE AND INSIDE LAPS, TRAVEL AND ANGULAR ADVANCE. (THURSTON.)

| CHANGE. | ADMISSION. | EXPANSION. | EXHAUST. | COMPRESSION. |
|---|---|---|---|---|
| Increase Outside Lap | begins later, ceases sooner | occurs earlier, continues longer | unchanged | unchanged |
| Decrease Outside Lap | begins earlier, ceases later | begins later, period shortened | unchanged | unchanged |
| Increase Inside Lap | unchanged | begins as before, continues longer | begins later, ceases earlier | begins sooner, contin. longer |
| Decrease Inside Lap | unchanged | begins as before, period shortened | begins earlier, ceases later | begins later. period short'd |
| Increase Travel | begins sooner, ceases later | begins later, ceases sooner | begins later, ceases later | begins later, ends sooner |
| Decrease Travel | begins later, ceases earlier | begins earlier, ceases later | begins earlier, ceases earlier | begins earlier, ceases later |
| Increase Angular Advance | begins earlier, period unchanged | begins sooner, period unchanged | begins earlier, period unchanged | begins earlier, period unchanged |
| Decrease Angular Advance | begins later, period unchanged | begins later, period unchanged | begins later, period unchanged | begins later, period unchanged |

## TABLE IV.

### PORT AREA, PORT WIDTH, AND STEAM-PIPE DIAMETER FOR VARIOUS PISTON SPEEDS AND STEAM VELOCITIES.

| | Velocity of Steam. Feet per Minute. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 4,000. | | | 6,000. | | | 8,000. | | | 10,000. | | | 12,000. | | |
| Piston Speed. Feet per Minute. | Port Area. Piston Area as Unity. | Steam-pipe Diameter. Piston Diameter as Unity. | Port Width. Piston Diameter as Unity. | Port Area. Piston Area as Unity. | Steam-pipe Diameter. Piston Diameter as Unity. | Port Width. Piston Diameter as Unity. | Port Area. Piston Area as Unity. | Steam-pipe Diameter. Piston Diameter as Unity. | Port Width. Piston Diameter as Unity. | Port Area. Piston Area as Unity. | Steam-pipe Diameter. Piston Diameter as Unity. | Port Width. Piston Diameter as Unity. | Port Area. Piston Area as Unity. | Steam-pipe Diameter. Piston Diameter as Unity. | Port Width. Piston Diameter as Unity. |
| 100 | .025 | .158 | .022 | .017 | .129 | .015 | .013 | .112 | .011 | .010 | .100 | .009 | .008 | .091 | .007 |
| 125 | .031 | .177 | .027 | .021 | .144 | .018 | .016 | .125 | .014 | .013 | .112 | .011 | .010 | .102 | .009 |
| 150 | .037 | .194 | .032 | .025 | .158 | .022 | .019 | .137 | .017 | .015 | .123 | .013 | .013 | .112 | .011 |
| 175 | .044 | .209 | .038 | .029 | .171 | .025 | .022 | .148 | .019 | .018 | .132 | .015 | .015 | .121 | .013 |
| 200 | .050 | .224 | .043 | .033 | .183 | .029 | .025 | .158 | .022 | .020 | .141 | .017 | .017 | .129 | .015 |
| 225 | .056 | .237 | .049 | .038 | .194 | .033 | .028 | .168 | .024 | .023 | .150 | .020 | .019 | .137 | .016 |
| 250 | .063 | .250 | .055 | .042 | .204 | .037 | .031 | .177 | .027 | .025 | .158 | .022 | .021 | .144 | .018 |
| 275 | .069 | .262 | .060 | .046 | .214 | .040 | .034 | .185 | .030 | .028 | .166 | .024 | .023 | .151 | .020 |
| 300 | .075 | .274 | .065 | .050 | .224 | .044 | .038 | .193 | .033 | .030 | .173 | .026 | .025 | .157 | .022 |
| 325 | .081 | .285 | .070 | .054 | .233 | .047 | .041 | .201 | .036 | .033 | .180 | .028 | .027 | .164 | .024 |
| 350 | .088 | .296 | .076 | .058 | .242 | .051 | .044 | .209 | .038 | .035 | .187 | .031 | .029 | .171 | .025 |
| 375 | .094 | .306 | .081 | .063 | .250 | .055 | .047 | .217 | .041 | .038 | .194 | .033 | .031 | .177 | .027 |
| 400 | .100 | .313 | .086 | .067 | .258 | .058 | .050 | .224 | .044 | .040 | .200 | .035 | .033 | .183 | .029 |
| 425 | .106 | .326 | .092 | .071 | .266 | .062 | .053 | .231 | .046 | .043 | .206 | .037 | .035 | .188 | .031 |
| 450 | .113 | .335 | .098 | .075 | .274 | .065 | .056 | .238 | .049 | .045 | .212 | .039 | .038 | .193 | .032 |
| 475 | .119 | .344 | .103 | .079 | .281 | .069 | .059 | .244 | .052 | .048 | .218 | .041 | .040 | .199 | .033 |
| 500 | .125 | .353 | .108 | .083 | .288 | .073 | .063 | .250 | .055 | .050 | .224 | .044 | .042 | .204 | .036 |
| 525 | .131 | .362 | .113 | .088 | .295 | .077 | .066 | .256 | .058 | .053 | .220 | .046 | .044 | .209 | .038 |
| 550 | .138 | .371 | .119 | .092 | .302 | .080 | .069 | .262 | .060 | .055 | .235 | .048 | .046 | .214 | .040 |
| 575 | .144 | .380 | .124 | .096 | .309 | .084 | .072 | .268 | .063 | .058 | .240 | .050 | .048 | .219 | .042 |
| 600 | .150 | .388 | .130 | .100 | .316 | .087 | .075 | .274 | .065 | .060 | .245 | .052 | .050 | .224 | .044 |
| 625 | .156 | .395 | .135 | .104 | .323 | .091 | .078 | .279 | .068 | .063 | .250 | .055 | .052 | .228 | .045 |
| 650 | .163 | .403 | .141 | .108 | .329 | .094 | .081 | .285 | .071 | .065 | .255 | .057 | .054 | .232 | .047 |
| 675 | .169 | .411 | .146 | .113 | .335 | .098 | .084 | .290 | .074 | .068 | .260 | .059 | .056 | .237 | .049 |
| 700 | .175 | .418 | .150 | .117 | .341 | .102 | .088 | .296 | .077 | .070 | .265 | .061 | .058 | .241 | .051 |
| 725 | .181 | .426 | .155 | .121 | .347 | .106 | .091 | .301 | .079 | .073 | .269 | .063 | .060 | .246 | .053 |
| 750 | .188 | .433 | .161 | .125 | .353 | .109 | .094 | .306 | .082 | .075 | .274 | .065 | .063 | .250 | .055 |
| 775 | .194 | .440 | .166 | .129 | .359 | .113 | .097 | .311 | .085 | .078 | .278 | .068 | .065 | .254 | .056 |
| 800 | .200 | .447 | .172 | .133 | .365 | .116 | .100 | .316 | .087 | .080 | .283 | .070 | .067 | .259 | .058 |
| 825 | .206 | .454 | .177 | .137 | .371 | .120 | .103 | .321 | .090 | .083 | .287 | .072 | .069 | .262 | .060 |
| 850 | .213 | .461 | .183 | .141 | .376 | .123 | .106 | .326 | .093 | .085 | .292 | .074 | .071 | .266 | .062 |
| 875 | .219 | .468 | .188 | .145 | .382 | .127 | .109 | .331 | .095 | .088 | .296 | .076 | .073 | .270 | .064 |
| 900 | .225 | .474 | .193 | .150 | .388 | .131 | .113 | .336 | .098 | .090 | .300 | .079 | .075 | .274 | .065 |
| 925 | .231 | .481 | .198 | .154 | .393 | .134 | .116 | .340 | .101 | .093 | .304 | .081 | .077 | .277 | .067 |
| 950 | .238 | .487 | .204 | .158 | .398 | .138 | .119 | .344 | .104 | .095 | .308 | .083 | .079 | .281 | .069 |
| 975 | .244 | .492 | .209 | .162 | .403 | .141 | .122 | .349 | .106 | .098 | .312 | .085 | .081 | .285 | .071 |
| 1000 | .250 | .500 | .214 | .166 | .408 | .145 | .125 | .353 | .109 | .100 | .316 | .087 | .083 | .289 | .073 |
| 1025 | .256 | .506 | .220 | .170 | .413 | .149 | .128 | .357 | .112 | .103 | .320 | .089 | .085 | .292 | .075 |
| 1050 | .263 | .512 | .225 | .175 | .418 | .153 | .131 | .361 | .114 | .105 | .324 | .092 | .088 | .295 | .076 |
| 1075 | .269 | .518 | .231 | .179 | .423 | .156 | .134 | .365 | .117 | .108 | .328 | .094 | .090 | .299 | .078 |
| 1100 | .275 | .524 | .236 | .183 | .428 | .160 | .138 | .370 | .120 | .110 | .332 | .096 | .092 | .303 | .080 |
| 1125 | .281 | .530 | .241 | .187 | .433 | .163 | .141 | .375 | .123 | .113 | .335 | .098 | .094 | .306 | .082 |
| 1150 | .288 | .536 | .246 | .191 | .438 | .167 | .144 | .379 | .126 | .115 | .339 | .100 | .096 | .310 | .084 |
| 1175 | .294 | .542 | .251 | .195 | .443 | .170 | .147 | .384 | .128 | .118 | .343 | .103 | .098 | .313 | .085 |
| 1200 | .300 | .548 | .256 | .200 | .447 | .175 | .150 | .388 | .131 | .120 | .346 | .105 | .100 | .316 | .087 |

# TABLE V.

## AREAS OF CIRCLES.

**Advancing by Eighths.**

| Diam. | Area. | Diam. | Area. | Diam. | Area. | Diam. | Area. |
|---|---|---|---|---|---|---|---|
| 1/64 | .00019 | 3 | 7.0686 | 1/8 | 51.849 | 1/4 | 207.39 |
| 1/32 | .00077 | 1/16 | 7.3662 | 1/4 | 53.456 | 3/8 | 210.60 |
| 3/64 | .00173 | 1/8 | 7.6699 | 3/8 | 55.088 | 1/2 | 213.82 |
| 1/16 | .00307 | 3/16 | 7.9798 | 1/2 | 56.745 | 5/8 | 217.08 |
| 3/32 | .00690 | 1/4 | 8.2958 | 5/8 | 58.426 | 3/4 | 220.35 |
| 1/8 | .01227 | 5/16 | 8.6179 | 3/4 | 60.132 | 7/8 | 223.65 |
| 5/32 | .01917 | 3/8 | 8.9462 | 7/8 | 61.862 | 17 | 226.98 |
| 3/16 | .02761 | 7/16 | 9.2806 | 9 | 63.617 | 1/8 | 230.33 |
| 7/32 | .03758 | 1/2 | 9.6211 | 1/8 | 65.397 | 1/4 | 233.71 |
| 1/4 | .04909 | 9/16 | 9.9678 | 1/4 | 67.201 | 3/8 | 237.10 |
| 9/32 | .06213 | 5/8 | 10.321 | 3/8 | 69.029 | 1/2 | 240.53 |
| 5/16 | .07670 | 11/16 | 10.680 | 1/2 | 70.882 | 5/8 | 243.98 |
| 11/32 | .09281 | 3/4 | 11.045 | 5/8 | 72.760 | 3/4 | 247.45 |
| 3/8 | .11045 | 13/16 | 11.416 | 3/4 | 74.662 | 7/8 | 250.95 |
| 13/32 | .12962 | 7/8 | 11.793 | 7/8 | 76.589 | 18 | 254.47 |
| 7/16 | .15033 | 15/16 | 12.177 | 10 | 78.540 | 1/8 | 258.02 |
| 15/32 | .17257 | 4 | 12.566 | 1/8 | 80.516 | 1/4 | 261.59 |
| 1/2 | .19635 | 1/16 | 12.962 | 1/4 | 82.516 | 3/8 | 265.18 |
| 17/32 | .22166 | 1/8 | 13.364 | 3/8 | 84.541 | 1/2 | 268.80 |
| 9/16 | .24850 | 3/16 | 13.772 | 1/2 | 86.590 | 5/8 | 272.45 |
| 19/32 | .27688 | 1/4 | 14.186 | 5/8 | 88.664 | 3/4 | 276.12 |
| 5/8 | .30680 | 5/16 | 14.607 | 3/4 | 90.763 | 7/8 | 279.81 |
| 21/32 | .33824 | 3/8 | 15.033 | 7/8 | 92.886 | 19 | 283.53 |
| 11/16 | .37122 | 7/16 | 15.466 | 11 | 95.033 | 1/8 | 287.27 |
| 23/32 | .40574 | 1/2 | 15.904 | 1/8 | 97.205 | 1/4 | 291.04 |
| 3/4 | .44179 | 9/16 | 16.349 | 1/4 | 99.402 | 3/8 | 294.83 |
| 25/32 | .47937 | 5/8 | 16.800 | 3/8 | 101.62 | 1/2 | 298.65 |
| 13/16 | .51849 | 11/16 | 17.257 | 1/2 | 103.87 | 5/8 | 302.49 |
| 27/32 | .55914 | 3/4 | 17.728 | 5/8 | 106.14 | 3/4 | 306.35 |
| 7/8 | .60132 | 13/16 | 18.190 | 3/4 | 108.43 | 7/8 | 310.24 |
| 29/32 | .64504 | 7/8 | 18.665 | 7/8 | 110.75 | 20 | 314.16 |
| 15/16 | .69029 | 15/16 | 19.147 | 12 | 113.10 | 1/8 | 318.10 |
| 31/32 | .73708 | 5 | 19.635 | 1/8 | 115.47 | 1/4 | 322.06 |
| 1 | .7854 | 1/16 | 20.129 | 1/4 | 117.86 | 3/8 | 326.05 |
| 1/16 | .8866 | 1/8 | 20.629 | 3/8 | 120.28 | 1/2 | 330.06 |
| 1/8 | .9940 | 3/16 | 21.135 | 1/2 | 122.72 | 5/8 | 334.10 |
| 3/16 | 1.1075 | 1/4 | 21.648 | 5/8 | 125.19 | 3/4 | 338.16 |
| 1/4 | 1.2272 | 5/16 | 22.166 | 3/4 | 127.68 | 7/8 | 342.25 |
| 5/16 | 1.3530 | 3/8 | 22.691 | 7/8 | 130.19 | 21 | 346.36 |
| 3/8 | 1.4849 | 7/16 | 23.221 | 13 | 132.73 | 1/8 | 350.50 |
| 7/16 | 1.6230 | 1/2 | 23.758 | 1/8 | 135.30 | 1/4 | 354.66 |
| 1/2 | 1.7671 | 9/16 | 24.301 | 1/4 | 137.89 | 3/8 | 358.84 |
| 9/16 | 1.9175 | 5/8 | 24.850 | 3/8 | 140.50 | 1/2 | 363.05 |
| 5/8 | 2.0739 | 11/16 | 25.406 | 1/2 | 143.14 | 5/8 | 367 28 |
| 11/16 | 2.2365 | 3/4 | 25.967 | 5/8 | 145.80 | 3/4 | 371.54 |
| 3/4 | 2.4053 | 13/16 | 26.535 | 3/4 | 148.49 | 7/8 | 375 83 |
| 13/16 | 2.5802 | 7/8 | 27.109 | 7/8 | 151.20 | 22 | 380.13 |
| 7/8 | 2.7612 | 15/16 | 27.688 | 14 | 153.94 | 1/8 | 384.46 |
| 15/16 | 2.9483 | 6 | 28.274 | 1/8 | 156.70 | 1/4 | 388.82 |
| 2 | 3.1416 | 1/8 | 29.465 | 1/4 | 159.48 | 3/8 | 393.20 |
| 1/16 | 3.3410 | 1/4 | 30.680 | 3/8 | 162.30 | 1/2 | 397.61 |
| 1/8 | 3.5466 | 3/8 | 31.919 | 1/2 | 165.13 | 5/8 | 402.04 |
| 3/16 | 3.7583 | 1/2 | 33.183 | 5/8 | 167.99 | 3/4 | 406.49 |
| 1/4 | 3.9761 | 5/8 | 34.472 | 3/4 | 170.87 | 7/8 | 410.97 |
| 5/16 | 4.2000 | 3/4 | 35.785 | 7/8 | 173.78 | 23 | 415.48 |
| 3/8 | 4.4301 | 7/8 | 37.122 | 15 | 176.71 | 1/8 | 420.00 |
| 7/16 | 4.6664 | 7 | 38.485 | 1/8 | 179.67 | 1/4 | 424.56 |
| 1/2 | 4.9087 | 1/8 | 39.871 | 1/4 | 182.65 | 3/8 | 429.13 |
| 9/16 | 5.1572 | 1/4 | 41.282 | 3/8 | 185.66 | 1/2 | 433.74 |
| 5/8 | 5.4119 | 3/8 | 42.718 | 1/2 | 188.69 | 5/8 | 438.36 |
| 11/16 | 5.6727 | 1/2 | 44.179 | 5/8 | 191.75 | 3/4 | 443.01 |
| 3/4 | 5.9396 | 5/8 | 45.664 | 3/4 | 194.83 | 7/8 | 447.69 |
| 13/16 | 6.2126 | 3/4 | 47.173 | 7/8 | 197.93 | 24 | 452.39 |
| 7/8 | 6.4918 | 7/8 | 48.707 | 16 | 201.06 | 1/8 | 457.11 |
| 15/16 | 6.7771 | 8 | 50.265 | 1/8 | 204.22 | 1/4 | 461.86 |

AREAS OF CIRCLES.

| Diam. | Area. | Diam. | Area. | Diam. | Area. | Diam. | Area. |
|---|---|---|---|---|---|---|---|
| ⅜ | 466.64 | 33 | 855.30 | ⅝ | 1360.8 | ¼ | 1983.2 |
| ½ | 471.44 | ⅛ | 861.79 | ¾ | 1369.0 | ⅜ | 1993.1 |
| ⅝ | 476.26 | ¼ | 868.31 | ⅞ | 1377.2 | ½ | 2003.0 |
| ¾ | 481.11 | ⅜ | 874.85 | 42 | 1385.4 | ⅝ | 2012.9 |
| ⅞ | 485.98 | ½ | 881.41 | ⅛ | 1393.7 | ¾ | 2022.8 |
| 25 | 490.87 | ⅝ | 888.00 | ¼ | 1402.0 | ⅞ | 2032.8 |
| ⅛ | 495.79 | ¾ | 894.62 | ⅜ | 1410.3 | 51 | 2042.8 |
| ¼ | 500.74 | ⅞ | 901.26 | ½ | 1418.6 | ⅛ | 2052.8 |
| ⅜ | 505.71 | 34 | 907.92 | ⅝ | 1427.0 | ¼ | 2062.9 |
| ½ | 510.71 | ⅛ | 914.61 | ¾ | 1435 4 | ⅜ | 2073.0 |
| ⅝ | 515.72 | ¼ | 921.32 | ⅞ | 1443 8 | ½ | 2083.1 |
| ¾ | 520.77 | ⅜ | 928.06 | 43 | 1452.2 | ⅝ | 2093.2 |
| ⅞ | 525.84 | ½ | 934.82 | ⅛ | 1460.7 | ¾ | 2103.3 |
| 26 | 330.93 | ⅝ | 941.61 | ¼ | 1469 1 | ⅞ | 2113.5 |
| ⅛ | 536.05 | ¾ | 948.42 | ⅜ | 1477.6 | 52 | 2123.7 |
| ¼ | 541.19 | ⅞ | 955.25 | ½ | 1486.2 | ⅛ | 2133.9 |
| ⅜ | 546.35 | 35 | 962.11 | ⅝ | 1494.7 | ¼ | 2144.2 |
| ½ | 551.55 | ⅛ | 969.00 | ¾ | 1503.3 | ⅜ | 2154.5 |
| ⅝ | 556.76 | ¼ | 975.91 | ⅞ | 1511.9 | ½ | 2164.8 |
| ¾ | 562.00 | ⅜ | 982.84 | 44 | 1520.5 | ⅝ | 2175.1 |
| ⅞ | 567.27 | ½ | 989 80 | ⅛ | 1529.2 | ¾ | 2185.4 |
| 27 | 572.56 | ⅝ | 996.78 | ¼ | 1537.9 | ⅞ | 2195.8 |
| ⅛ | 577.87 | ¾ | 1003.8 | ⅜ | 1546.6 | 53 | 2206.2 |
| ¼ | 583.21 | ⅞ | 1010.8 | ½ | 1555.3 | ⅛ | 2216.6 |
| ⅜ | 588.57 | 36 | 1017.9 | ⅝ | 1564.0 | ¼ | 2227.0 |
| ½ | 593.96 | ⅛ | 1025.0 | ¾ | 1572.8 | ⅜ | 2237.5 |
| ⅝ | 599.37 | ¼ | 1032.1 | ⅞ | 1581.6 | ½ | 2248.0 |
| ¾ | 604.81 | ⅜ | 1039.2 | 45 | 1590.4 | ⅝ | 2258.5 |
| ⅞ | 610.27 | ½ | 1046.3 | ⅛ | 1599.3 | ¾ | 2269.1 |
| 28 | 615.75 | ⅝ | 1053.5 | ¼ | 1608.2 | ⅞ | 2279.6 |
| ⅛ | 621.26 | ¾ | 1060.7 | ⅜ | 1617.0 | 54 | 2290.2 |
| ¼ | 626.80 | ⅞ | 1068.0 | ½ | 1626.0 | ⅛ | 2300.8 |
| ⅜ | 632.36 | 37 | 1075.2 | ⅝ | 1634.9 | ¼ | 2311.5 |
| ½ | 637.94 | ⅛ | 1082.5 | ¾ | 1643.9 | ⅜ | 2322.1 |
| ⅝ | 643.55 | ¼ | 1089.8 | ⅞ | 1652.9 | ½ | 2332.8 |
| ¾ | 649.18 | ⅜ | 1097.1 | 46 | 1661.9 | ⅝ | 2343.5 |
| ⅞ | 654.84 | ½ | 1104.5 | ⅛ | 1670.9 | ¾ | 2354.3 |
| 29 | 660.52 | ⅝ | 1111.8 | ¼ | 1680.0 | ⅞ | 2365.0 |
| ⅛ | 666.23 | ¾ | 1119.2 | ⅜ | 1689.1 | 55 | 2375.8 |
| ¼ | 671.96 | ⅞ | 1126.7 | ½ | 1698.2 | ⅛ | 2386.6 |
| ⅜ | 677.71 | 38 | 1134.1 | ⅝ | 1707.4 | ¼ | 2397.5 |
| ½ | 683.49 | ⅛ | 1141.6 | ¾ | 1716.5 | ⅜ | 2408.3 |
| ⅝ | 689.30 | ¼ | 1149.1 | ⅞ | 1725.7 | ½ | 2419.2 |
| ¾ | 695.13 | ⅜ | 1156.6 | 47 | 1734.9 | ⅝ | 2430.1 |
| ⅞ | 700.98 | ½ | 1164.2 | ⅛ | 1744.2 | ¾ | 2441.1 |
| 30 | 706.86 | ⅝ | 1171.7 | ¼ | 1753.5 | ⅞ | 2452.0 |
| ⅛ | 712.76 | ¾ | 1179.3 | ⅜ | 1762.7 | 56 | 2463.0 |
| ¼ | 718.69 | ⅞ | 1186.9 | ½ | 1772.1 | ⅛ | 2474 0 |
| ⅜ | 724.64 | 39 | 1194.6 | ⅝ | 1781.4 | ¼ | 2485.0 |
| ½ | 730.62 | ⅛ | 1202.3 | ¾ | 1790.8 | ⅜ | 2496.1 |
| ⅝ | 736.62 | ¼ | 1210.0 | ⅞ | 1800.1 | ½ | 2507.2 |
| ¾ | 742.64 | ⅜ | 1217.7 | 48 | 1809.6 | ⅝ | 2518.3 |
| ⅞ | 748.69 | ½ | 1225.4 | ⅛ | 1819.0 | ¾ | 2529.4 |
| 31 | 754.77 | ⅝ | 1233.2 | ¼ | 1828 5 | ⅞ | 2540.6 |
| ⅛ | 760.87 | ¾ | 1241.0 | ⅜ | 1837.9 | 57 | 2551.8 |
| ¼ | 766.99 | ⅞ | 1248.8 | ½ | 1847.5 | ⅛ | 2563.0 |
| ⅜ | 773.14 | 40 | 1256.6 | ⅝ | 1857.0 | ¼ | 2574.2 |
| ½ | 779 31 | ⅛ | 1264.5 | ¾ | 1866.5 | ⅜ | 2585.4 |
| ⅝ | 785.51 | ¼ | 1272.4 | ⅞ | 1876.1 | ½ | 2596.7 |
| ¾ | 791.73 | ⅜ | 1280.3 | 49 | 1885.7 | ⅝ | 2608.0 |
| ⅞ | 797.98 | ½ | 1288.2 | ⅛ | 1805.4 | ¾ | 2619.4 |
| 32 | 804.25 | ⅝ | 1296.2 | ¼ | 1905.0 | ⅞ | 2630.7 |
| ⅛ | 810.54 | ¾ | 1304.2 | ⅜ | 1914.7 | 58 | 2642.1 |
| ¼ | 816.86 | ⅞ | 1312.2 | ½ | 1924.4 | ⅛ | 2653.5 |
| ⅜ | 823.21 | 41 | 1320.3 | ⅝ | 1934.2 | ¼ | 2664.9 |
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